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THE BRITISH SHIP RESEARCH ASSOCIATION

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ABSTRACTS *from Current Technical Literature*

The following Abstracts purport to be fair summaries of the articles, but the Association does not accept responsibility for statements made in the originals, nor does it necessarily agree with their contents

The standard form of reference to the source of each Abstract is: Title of Periodical or Publication (abbreviated according to the list on pp. 3–19 of B.S.R.A. Journal for January 1968), volume number (in heavy type), year, and page number, followed by the date of issue where appropriate. The length of the article and other bibliographical details are also indicated.

SHIP RESISTANCE AND FLUID MOTION

26,519 **A Major Experiment in Ship Research** (in German). SCHUSTER, S. *V.D.I.Z.*, **109**, No. 32 (1967), p. 1516 (Nov.) [2 pp., 3 phot.]

In the latter part of 1967, the research-ship *Meteor* (see Abstract No 24,048, Mar. 1966) was equipped with three slewable aircraft jet engines. (mounted on a bridge over the stern ramp) for propelling the ship during resistance and other tests as part of a study of ship/model correlation. These engines provided a maximum thrust of about 10 tons, sufficient for a speed of about 12 knots. Complementing the resistance tests, propulsion tests were carried out using the ship's normal propulsion machinery at 12 knots (about 2,500 h.p.; thrust about 12 tons). Course-stability and manœuvrability tests were also made. The measurements taken (and automatically recorded) included velocity distributions in the vicinity of the propeller, boundary-layer velocities at two positions on the shell, rudder angles and forces, wind speed and direction, and airborne and structural noise.

The VWS (W. Berlin Tank), the HSVA (Hamburg Tank), and the I.f.S. (of Hamburg University), three organisations who collaborated in this work, are making separate analyses of the results and relating them to the results of model tests (including tests on families of models). Conclusions are expected concerning scale effect on thrust deduction fraction and wake fraction, nominal and effective wake distribution, hydrodynamic vibration-excitation at the propeller, and course-stability and manœuvrability characteristics. The results of this evaluation may become available in 1968.

The article includes a brief history of previous resistance and associated measurements on actual ships or very large (boat-like) models (none have been on a ship as large as the *Meteor*), undertaken to investigate ship/model correlation. Mention is made of B.S.R.A.'s work (in 1950–51) with *Lucy Ashton*, which was likewise driven by jet engines. The results, as a whole, have been inconsistent, possibly because of inadequate measuring equipment. In the present investigation, therefore, great care was taken to ensure satisfactory equipment, an adequate number of trained personnel, and sufficient time for the tests.

PROPELLERS AND PROPULSION

(*See also Abstract No.* 26,519)

26,520 **Efficiency of Shrouded Propellers with Various Nozzle and Propeller Forms** (in German). SCHROEDER, G. *Schiffbautechnik*, **17** (1967), p. 428 (Aug.) [4 pp., 11 ref., 1 tab., 6 graphs, 5 diag.]

This is Publication No. 44 of the Schiffbau-Versuchsanstalt, Potsdam, East Germany.

This article is a sequel to those covered by Abstract No. 25,312 (May 1967); the results are presented and discussed of further open-water model tests on shrouded propellers. The objects of the present work were (*a*) direct experimental comparison between Wageningen-type and the SVA's simplified (Shushkin-type) nozzles, (*b*) a study of the influence of exit-area ratio on efficiency, (*c*) investigation of possible efficiency reductions in lightly-loaded nozzle propellers, (*d*) comparison between a conventional (B3.35) and a Kaplan (Ka 4.55) screw in both types of nozzle. In all, nine different nozzle designs were tested.

Various detailed conclusions are drawn. It is found, *inter alia*, that an increase in exit-area ratio considerably improves " astern " efficiency and has no adverse effect on ahead efficiency. The trailing-edge radius of the nozzle profile should not exceed 0·02 times the nozzle radius at the propeller disc. An SVA nozzle was superior to a Wageningen nozzle at low loadings; at such loadings the choice of a suitable screw is particularly important. Although model results indicate a slight advantage of Kaplan over conventional screws at high loadings, this is not apparent in full-scale " static " tests.

26,521 **Shrouded Propellers for Tankers—An Investigation into the Costs.** *Shipp. World & Shipb.*, **160** (1967), p. 2079 (Dec.) [3 pp., 8 ref., 1 tab., 6 diag.]

No information appears to be available on the method and cost of construction of propeller shrouds for very large tankers and bulk carriers, although the costs (construction, installation, and maintenance) of these nozzles would be an important factor in deciding whether to incorporate one in a new ship. The fixed type of nozzle would probably be slightly cheaper to construct than a nozzle rudder but would be more expensive to install; maintenance costs would be about the same for each type. A nozzle rudder would allow a more efficient stern design from the propulsion standpoint, and would permit easier removal of the propeller and withdrawal of the tailshaft; it would, however, have less structural rigidity than a fixed nozzle. These factors were taken into consideration by the (anonymous) Author in developing a structural design for a nozzle rudder, in preference to one for a fixed nozzle, for large ships. The design is based on an outline drawing of a Kort nozzle rudder, given in an article by J. N. Wood published in 1966; the nozzle profile is derived from the Wageningen series charts published by Van Manen in the 1950s (see Abstract No. 15,264, May 1959).

The nozzle, of 28·2 ft inside diameter, is of welded cellular construction, the " base " being a heavy steel plate (2 in thick in the example described) forming a cylinder around the path of the propeller-blade tips. The remainder of the plating forming the profile is much thinner; the only parts of it having double curvature (circular arcs in the profile plane) are

those ahead of the base plate. The leading and trailing edges are solid-drawn steel pipes bent into a circular form; these are the main transverse members. There are internal stiffeners, both longitudinal and circumferential. These and other details of the nozzle structure, including its "rudder stock", are described and shown in dimensioned drawings. The stock torque is assumed to be 300 tons-ft. The strength calculations are briefly explained, and a list of scantlings is given (there are no relevant Rules or published information on strength considerations; the Author uses simple flexural-rigidity theory as far as possible for the plating and stiffeners, and simple bending and torsion theory for the rudder stock).

Procedures for fabrication and testing of the nozzle rudder (which is estimated to weigh 50 tons), and for fitting it, are suggested; an assembly and welding sequence minimising distortion is outlined. It is estimated that construction would need 5,000 man-hours, and testing and painting another 200 man-hours; fitting (including tailshaft and propeller) would require 450 man-hours.

A nozzle rudder of this type should cost under £14,000 to construct and install. Maintenance costs would be limited to annual replacement of zinc anodes, and painting.

26,522 **The Voith "Wassertrecker" (Water Tractor), with Voith-Schneider Propulsion Installed Forward** (in German). BAER, W. *V.D.I.Z.*, **109**, No. 32 (1967), p. 1509 (Nov.) [4 pp., 6 ref., 4 diag., 1 phot.]

The Author briefly discusses the drawbacks of the conventionally-propelled Diesel harbour-tug. They led, in 1950, to the idea of developing a new type of harbour tug having the following features: (i) Voith-Schneider propulsion and steering, (ii) forward mounting of the Voith-Schneider propeller(s), which are protected below by plate(s) of special hydrodynamic form (" nozzle plates ") to reduce contraction losses and strong enough to support the full weight of the craft in dock, (iii) towing-hook fitted well aft, and (iv) a large vertical fin at the stern (to move the centre of lateral resistance aft).

The Author then discusses the advantages of the Voith " Wassertrecker " (Water Tractor) type of harbour tug, which has the features mentioned. General-arrangement drawings of a twin-propeller Wassertrecker are given. There follows an examination of the hydrodynamic problems involved in forward mounting of the propulsion/steering device, and their solution in the Wassertrecker.

The assessment of tugs by their horsepower or towing-pull is unsatisfactory. As its propellers can develop full thrust in any direction, the Wassertrecker needs less installed power than a screw-propelled tug engaged in the same ship-handling duties.

See also Abstracts No. 19,738 (Apr. 1963), and 22,259 (Oct. 1964).

26,523 **Some Experimental Results of Tests of a Low-Speed, Waterjet Propulsion System.** DELAO, M. *Journal of Hydronautics*, 1 (1967), p. 97 (Oct.) [4½ pp., 6 ref., 1 tab., 7 graphs, 1 diag., 3 phot.]

This is a revised version of a paper presented at the AIAA/USN Second Marine Systems and ASW Conference, Los Angeles/Long Beach, 8–10 Aug. 1966.

The Author presents some experimental data on the static and dynamic performance of a 16·5-in rotor-diameter waterjet propulsion system.

The 175-h.p., 2,500-r.p.m., direct-drive system was designed to propel a hydrofoil craft at 9 knots in the displacement condition. The required characteristics of the single-stage axial-flow pump are described and the performance results discussed. Methods of improving overall system efficiency by using more sophisticated hydrodynamic theory are suggested.

26,524 **Electromagnetic Propulsion for Cargo Submarines.** WAY, S. *Journal of Hydronautics*, 2 (1968), p. 49 (Apr.) [9 pp., 11 ref., 9 tab., 4 graphs, 15 diag., 1 phot.]

This is a revised version of Paper No. 67–363, presented at the AIAA/Soc. N.A.M.E. Advance Marine Vehicles Meeting, 22–24 May 1967.

Various schemes of electromagnetic propulsion, in which use is made of the electrical conductivity of sea water, have been proposed by the Author and other workers (see Abstracts No. 20,896, Jan. 1964, and 23,179, July 1965). These schemes can be categorised as D.C. (using steady, crossed electric and magnetic fields) or A.C. (using the interaction of a travelling magnetic field with the currents induced by it in the water); also, according to whether the thrust is generated in an internal longitudinal duct or in the external flow around the hull. After briefly reviewing the relative merits of these arrangements, the Author selects the external-field D.C. system for further study.

He explains at some length why the application of electromagnetic propulsion to large submarine tankers may lead to increased propulsion efficiency. The general theory of external-field D.C. propulsion is discussed, and propulsion efficiencies are deduced for 2-, 4-, and 6-pole configurations. Application to submarine-tanker hulls of length/diameter ratio 8·75 and prismatic coefficient 0·68 is then discussed for craft with 4 or 6 poles and submerged displacements of 25,000, 50,000, and 100,000 metric tons. For the 6-pole arrangement, at 29 knots, the thrust power is estimated to be 86% of the electric power supplied at 100,000 tons, 83% at 50,000 tons, and 79% at 25,000 tons. Higher values (over 90%) are obtained at 20 knots.

The arrangement of equipment inside the hull is discussed. Attention must be given to supporting the coils of the superconductor magnets; separating forces are very large and the restraints must not give excessive heat leakage. The refrigerating power requirement must be kept below about 5% of the full-load power. Other problems are those of the magnetic field inside the hull, and the attraction of foreign iron bodies.

Some particulars are given of a 10-ft fibreglass model submarine (provided with batteries and external-field conductors of ordinary materials) which has been successfully test-run at the University of California.

SHIP PERFORMANCE, STABILITY, AND MANOEUVRABILITY

(See also Abstracts No. 26,519 and 26,534)

26,525 **Comparative Results of Trials on Sea-Going Timber-Carriers of 5,000 tons Cargo Deadweight** (in Russian). ZAKHAROV, B. N., and SHMELEV, A. V. *Sudostroenie*, No. 6 (1967), p. 3 (June) [6 pp., 4 ref., 6 tab., 3 graphs, 3 phot.]

This article gives the main results of trials carried out in 1961–1964

on three timber-carriers, *viz* the motor ships *Vytegrales* and *Belomorskles*, and the free-piston gas-turbine vessel *Pavlin Vinogradov* (this ship is described in Abstracts No. 17,475, June 1961, and 19,030, Oct. 1962; see also Abstract No. 21,553, June 1964). Timber-carriers of the *Vytegrales* and *Belomorskles* classes continue to be built in the U.S.S.R. and Poland, respectively, and six Soviet-built *Pavlin Vinogradov*-class ships are being successfully operated. They all have four holds. The superstructure and machinery are three-quarters aft (between holds Nos 3 and 4) on *Vytegrales*, amidships on *Belomorskles*, and aft on *Pavlin Vinogradov*. The relative merits of these arrangements as regards stowage of deck cargo are discussed. Tables give for each ship: (*a*) a breakdown of deck area according to nature of utilisation or obstruction; (*b*) the aggregate cubic capacities in categories such as holds, ballast tanks, and bunkers, and also the total underdeck volume; (*c*) the cargo deadweights, quantities of ballast, and stability characteristics in specified conditions, light weight and hull c.g., and the utilisation of displacement. The differences between the three ships in these respects are discussed.

The results of speed trials are compared graphically (r.p.m. and power curves). It can be deduced that, at the same draught and machinery output, the " full ahead " speed of *Belomorskles* is 0·45 knots higher in the loaded condition and 0·50 knots higher in the ballast condition than that of *Vytegrales*; this is attributed partly to the lower efficiency of the faster-running propeller in *Vytegrales*, and partly to her higher block coefficient. The trials of *Pavlin Vinogradov* showed that the specified speed of 14 knots could be achieved only after fitting a propeller of reduced pitch; this was done (see Abstract No. 21,553).

Those dimensional characteristics of the holds and hatches of the ships which influence the rate of cargo-handling are summarised in a table. It appears that these ships are not sufficiently " open "; in the recent *Sibirles* class (see Abstract No. 24,872, Dec. 1966) the proportion of hold volume directly below the hatches is much greater. Data on the distribution of cubic capacity among the different holds are also given and discussed, with special reference to the end holds. The transverse side framing causes considerable inconvenience and loss of space.

The main engine of *Vytegrales* is a 9DKRN50/110 Diesel rated for 5,200 h.p. at 170 r.p.m., and that of *Belomorskles* is a B. & W. 562-VT2BF-140 rated for 5,450 h.p. at 135 r.p.m. The *Pavlin Vinogradov* has four SIGMA free-piston gas-generators feeding a power turbine rated for 4,000 h.p. at 115 propeller r.p.m. A table gives some particulars of these machinery installations; they include weights, the lengths and floor areas and volumes of the engine rooms, and engine-room area and volume utilisation factors (i.e. area or volume divided by b.h.p.). The free-piston installation has substantially higher utilisation factors than the others, but its weight advantage is cancelled by the fact that it consumes 10–20% more fuel.

On the basis of ship-trials results in the loaded condition, the principal performance characteristics of the three machinery installations are compared graphically. The Diesel engines are overloaded at the rated r.p.m. The B. & W. engine is the more efficient. The efficiency of the *Pavlin Vinogradov* machinery at intermediate and low speeds is unsatisfactory when all four gas generators are working. When navigational circumstances allow, the simultaneous use of all four generators should

be avoided. Test runs on heavy fuel were made (see also Abstract No. 19,439, Jan. 1963); they showed that the use of a heavier grade (also suitable for the low-speed Diesel installations) was possible and desirable.

Trials of the auxiliary Diesel engines showed that they were satisfactory; they are, however, often used in a way conducive to poor efficiency (e.g. two sets running simultaneously when one would cover the load). A table lists the ancillaries (and their rated throughputs) serving the main and auxiliary engines in each ship.

The trials and service experience showed that, in these timber carriers, the consumption of fresh water was two to three times greater than the amount required by the relevant sanitary standards; this was probably because the taps were not of self-closing type. Some particulars are given of the evaporators fitted in these ships; that in *Belomorskles* is of 2·5 tons/day capacity and is heated by the engine cooling water, whereas those in *Vytegrales* and *Pavlin Vinogradov*, of 3 and 10 tons/day respectively, use steam.

Typical 24-hr "at sea" load curves for the electric generating plant of each ship are given and discussed. In *Vytegrales* and *Pavlin Vinogradov* it is usual to have two generators on line at 35–50% load. The dynamic characteristics of the generators are unsatisfactory, because voltage drops up to 35% occur on starting large consumers, the corresponding recovery time being 6 sec. Self-excited generators with much better characteristics are now being installed in ships of the *Vytegrales* class.

26,526 **Three-Blade Rudders on Seagoing Ships** (in German). JENCKEL, F.-W. *Schiffbautechnik*, 17 (1967), p. 694 (Dec.) [3 pp., 4 ref., 5 diag., 2 phot.]

The Jenckel three-rudder system, and its application to inland-waterways and harbour craft, have previously been described (see Abstract No. 24,959, Jan. 1967). Its satisfactory performance led to its being considered as a means of improving the manœuvrability of East German stern trawlers, in particular that of a series of freezer trawlers (having a length b.p. of 44·3 m or 145 ft, a displacement of 844 cu m or 29,800 cu ft, and a service speed of 12 knots) building at the VEB Elbe-Werft, Boizenburg. The first few trawlers of this type had a nozzle rudder; this did not give satisfactory manœuvrability, particularly during trawling in heavy seas and side-winds, and the fifth and later ships of the series were fitted with a Jenckel three-rudder system behind a fixed Kort nozzle. A controllable-pitch propeller has been fitted from the outset.

This rudder and nozzle arrangement, adopted after tank tests had been quickly carried out, is giving satisfaction. The manœuvrability is now as good as that of a ship with an active rudder. The advantages due to the fixed Kort nozzle, and those due to the Jenckel rudders, are listed. The arrangement is shown in drawings, and is described and discussed. Among its features are: (*a*) near the trailing edge, the streamline section of each rudder changes to a short parallel section with a rounded trailing edge, (*b*) the bottom of each rudder is supported in a stout bearing [illegible] by a bar projecting aft from the nozzle, (*c*) the rudders are easily fitted and removed, (*d*) the arrangement does not allow fouling by nets or wires, (*e*) the rudders and nozzle have their hollow parts filled with pitch, and (*f*) the steering gear is a standard machine having a maximum torque of

4 tonne-metres or 13·1 tons-ft (this machine is used to only about 65% of its capacity).

A development of the system for small seagoing ships of up to about 3,000 h.p., such as stern trawlers, factory trawlers, fisheries-research ships, and coasters, is shown diagrammatically (a suitable bridge layout being indicated) and described; it includes an auto-pilot. The features and advantages of such a system are discussed. It would be particularly favourable in the case of combined inland-waterways/seagoing ships such as those using the Rhine or Danube. It is mentioned that the advantages of the Jenckel rudder (both with and without a Kort nozzle) for seagoing ships have been demonstrated in model tests carried out in Denmark for the Icelandic authorities.

26,527 **Underwater Hovering Control [by both Hydrostatic and Hydrodynamic Means] with Fluidic Amplifiers.** GOLDSCHMIED, F. R. *Journal of Hydronautics*, 2 (1968), p. 102 (Apr.) [6 pp., 12 ref., 9 graphs, 5 diag., 5 phot.]

This is a revised version of AIAA Paper No. 67–433, presented at Washington, D.C., 17–21 July 1967.

See also Abstract No. 26,511 (June 1968).

STRUCTURAL DESIGN AND ITS APPLICATIONS

(*See also Abstract No.* 26,535)

26,528 **Reflections upon Permissible Longitudinal Stresses in Ships.** STENEROTH, E. R. *R.I.N.A., Paper No. W10* (1966), *issued for written discussion* [7 pp., 23 ref., 6 graphs, 1 diag.]

The Author first briefly reviews present knowledge of statistical wave-load distributions. He then gives some values for hull-girder design stresses (wave and still-water) applicable to cargo liners and tankers, and explains why a relatively small allowance is made for slamming stresses. With these design stresses, normal ship steel gives an appreciable margin of safety against extensive yielding and an even greater one against collapse, provided precautions are taken to prevent buckling (as can readily be done). The failure records clearly show that, at present, the factor limiting nominal hull-girder design stresses is the occurrence of cracks. The cracks concerned are of two kinds—brittle and fatigue. It is often considered that fatigue cracks are not critical because they can be detected and repaired in good time, although they may aggravate a situation already favourable to brittle fracture, which should be the main risk determining permissible stress. The Author believes, however, that the best criterion is the stress level needed to form a visible crack, whether this propagates immediately as a brittle fracture or gradually as a fatigue crack. (Minute non-propagating fatigue cracks, less than 0·006 in long, can be expected at many points in a ship structure; reference is made to a study of these published by Frost and Dugdale in 1957.)

It seems practicable to design, and to manufacture in a shipyard, hull-structure components which are able to yield locally (under static loading) to such an extent that the nominal stress level in the whole structure reaches the yield-stress level without formation of a crack. More full-scale tests (like those published by J. Vasta—see Abstract No. 14,881,

Jan. 1959) on conventional structural details are needed to establish the present overall nominal-stress levels corresponding to initial failure, and to suggest improvements. Multi-axial states of stress and very severe stress concentrations do not correspond to a normal and acceptable standard of workmanship, and should not therefore limit the allowable stress. With normal details and ship steels, full safety against brittle-fracture initiation cannot be achieved merely by a limit on nominal stress. A possible mechanism for initiation of a brittle fracture at the front of a fatigue crack is suggested; it implies that initiation is possible at temperatures some degrees above the transition temperature, which may itself be raised some degrees by the fatigue mechanism.

Little is known about typical fatigue endurance curves for real ship structures, although some full-scale components have been tested (Nibbering's work, covered by Abstract No. 21,902, Sept. 1964, is mentioned). Most conclusions have been drawn from laboratory test specimens. The adverse effect of corrosion is noted; it is much reduced by normal corrosion-protection measures. On the basis of available experimental data, two "typical" sets of endurance curves (using the "visible-crack" criterion) for a structure are presented; one set corresponds to "bad" and the other to "good" details. Each curve corresponds to a different tensile mean stress. If the details are good, mean stress has very little effect on the short-life portion of the endurance curves. At long lives its effect is much the same for "good" and "bad" details; the effect of a given increment in mean stress is much less at high than at low mean stresses. These curves are discussed in the light of practical experience of cracking.

A comparison is made between tentative endurance curves for ship structures in mild and in higher-tensile steel, with "bad but acceptable" and with "good" details and welds. These show that, with good details, the "higher-tensile" curve can be higher than the "mild-steel" one even at long lives. The critical factor is commonly thought to be the fatigue strength of the welds. It is known that the geometrical effect of the weld contour is very important in relation to fatigue strength; recent tests by the Swedish Steelmakers' Association have shown that the strength properties of the electrode used also have an important influence. S–N curves show the results from manually butt-welded 20-mm (0·79 in) plate specimens with the same weld contour but made with different electrodes. They indicate the need to develop suitable electrodes to match the higher-strength steels, if the fatigue strength of welds is considered critical for a rise in permissible stress level. Given such electrodes, it seems likely that the endurance curve can be considerably higher for a structure in higher-strength steel than for a conventional structure. A large increase in nominal-stress levels would then be possible without increasing the risk of fatigue cracking.

Practical higher-strength steels (in particular, those micro-alloyed with niobium or vanadium) are now available with transition temperatures below −40° C (−40° F). With such steels catastrophic brittle failure does not seem possible, unless fatigue cracks have propagated so far that the remaining cross-sectional area of the hull girder is very small (a non-practical case).

The Author mentions the differences in the Rules of the various Classification Societies regarding permissible stress levels when higher-

strength steels are used. For short lives the ultimate strength of the steel is of little practical interest for a structure containing stress concentrations, but for long lives the fatigue strength of the structure will be determined by those strength properties of the steel which determine the endurance strength of the components; it is still doubtful whether ultimate strength or yield strength is the governing property. At present, attention to the design and fabrication of structural details is of greater practical importance. It seems justifiable to base the permissible nominal stress level mainly on the yield point of the steel. A rudimentary analysis indicates that slamming bending-stresses decrease as the L/D-ratio increases; if this is true the L/D ratio need not limit the utilisation of higher-strength steels in ships.

For a realistic approach to the longitudinal-strength problem, it is essential to carry out static tests on full-scale components. Until enough reliable results are available it will be inadvisable to increase the permissible stress-levels in ship structures, although good details would allow such an increase if the risk of brittle fracture could be eliminated. In certain higher-strength steels this risk is negligible, and stress levels even higher than those now allowed may be permissible after careful consideration of the fatigue characteristics.

26,529 **Optimum Design of Plates with Symmetrical Trapezoidal Corrugations Subjected to Lateral Pressure.** BASU, A. K., and CHAPMAN, J. C. *R.I.N.A., Paper No. W8* (1966), *issued for written discussion* [9 pp., 3 ref., 2 tab., 4 graphs, 3 diag.]

A method is proposed for the minimum-weight design of plates with symmetrical trapezoidal corrugations, subjected to lateral pressure. An equivalent stress (compounded from the three in-plane stress components, *viz* longitudinal, transverse, and shear) is calculated by the theory of von Mises–Huber–Hencky, which makes use of the shear-strain energy yield criterion. The individual stress components are calculated in the usual manner by the application of beam theory to isolated trough units. The magnitudes of these components are treated as variable while optimising the trough section for minimum weight, the permissible equivalent stress remaining constant.

The method simultaneously takes into account the effects of longitudinal and local transverse bending-moment, and also allows the provision of any desired factor of safety against buckling of the compression flange. Curves are given showing the variation of the optimum corrugation profile with the lateral pressure, and, to facilitate selection of actual trough dimensions, a set of design curves is provided in non-dimensional form.

To illustrate the variation of optimum trough-dimensions, and of weight per unit area of a corrugated plate, with span and external loading, numerical computations have been made for eight different cases; the results are tabulated and discussed. In all these cases the mild-steel plates are considered to be simply supported, subjected to uniform lateral pressure, and of sufficient width for the theory to be valid. A rigorous optimisation for spans with end fixity seems impracticable, but a tentative design procedure is suggested.

The possibilities of non-symmetrical trapezoidal corrugations are

briefly considered; these might be more efficient under conditions where the pressure is unidirectional and sufficiently small for buckling to be a criterion, but would be difficult to optimise without some restriction on trough pitch. Determination of the most efficient trough shape, without geometrical restriction, is also a very difficult problem.

The work described was part of an investigation, sponsored by B.S.R.A., on swedged and corrugated bulkheads.

26,530 **The Torsional Behaviour of Ships with Large Hatch Openings: Some Further Experiments.** WILDE, G. DE. *Shipp. World & Shipb.*, **161** (1968), p. 59 (Jan.) [6 pp., 3 ref., 8 graphs, 14 diag., 1 phot.]

The Author refers to a previous article (see Abstract No. 25,471, July 1967) in which he described a method of predicting the torsional behaviour of large-hatch ships which was based on the results of experiments with box-shaped models; the simple theory used in these experiments could only be considered valid for beams of constant cross-section. Further experiments on a model of ship-like form (but symmetrical fore and aft) were therefore arranged in order to investigate the influence of such form on end constraint and general behaviour of the beam. The present article describes these model experiments. Some of the information in the previous article is repeated for clarification.

The dimensions of the transparent p.v.c. model used are shown in a diagram; its length (6 ft) and midship section were the same as those of the prismatic model previously tested, and it had a single very large hatch. The experiments were carried out under constant torque in two conditions, the first without and the second with an inner skin. No variations in the size of hatch opening were investigated. The test rig, which was that used in the previous experiments with prismatic models, is shown in a photograph.

The angle of twist at a number of sections was measured, as were the normal stresses at two stations (both in way of the hatch but one close to its end). The results are shown graphically. Comparison of these results with those of the prismatic-model tests showed that the ship-like " shape " produced a large increase in the angle of twist and a reduction in end constraint. Strain readings taken only on the outer skin showed a pattern which differed fundamentally from that obtained for the prismatic model; instead of a reversal of stress over the depth, at the hatch-end station one side was completely in compression and the other in tension. The pattern at the other station was more normal. However, when tests were repeated with the inner skin also strain-gauged at the hatch-end station, the mean between the outer and inner skin measurements produced almost a straight line, which is in accordance with the general pattern predicted by torsion theory.

The theory presented assumes that the model behaviour can be predicted by using torsion theory for a prismatic beam having the same section properties as the midship section, when the correct boundary conditions and a shape correction factor (depending on fullness) are used.

The effect of shape on end constraint is discussed, and a theoretical method of determining this constraint is developed. It was found that, for the prediction of angles of twist, there was no need to introduce a correction factor depending on fullness. In the case of stress prediction, however, a correction factor of 1·5 was required to bring the line of

average stress for outer and inner skin into reasonable conformity with the measured results. Even this did not adequately predict the experimental stress pattern, because stresses due to secondary bending of the sides had not been taken into account. In the model, secondary stressing of the outer and inner skin arises from horizontal bending of the double sides, considering port and starboard as independent beams. An expression is therefore derived for this stress, which, on the inner skin, has to be added to and, on the outer skin, subtracted from the component arising from torsion. Stresses calculated in this way give better results but agreement is still not satisfactory, due partly to warping stresses in the skins caused by warping of the closed parts of the cross-section, and partly to secondary effects introduced by a change in cross-sectional shape. A similar type of stress distribution was found during re-testing of the prismatic model which had been previously gauged on the outside only, but agreement was better than in the shaped model.

The Author emphasises that, in the derivation of the differential equation for torsion, only shear stresses compatible with the primary system of normal stresses were taken into account. The shear stresses compatible with the secondary bending stresses are smaller, and the components in the outer and inner skins are in opposite directions, so that their contribution to the torque is determined by the spacing of the skins.

On the basis of the theory outlined, a computer program was developed which will calculate the equivalent deck thickness, torsional properties of the idealised sections, applied torque and the resulting angles of twist, normal stresses, and shear stresses. By this means the influence of a number of factors on stresses and deflections was calculated. The results of these calculations are shown graphically, and discussed under: (*a*) Influence of (ship) size. (*b*) Influence of hatch length. (*c*) Influence of hatch width. (*d*) Influence of width of structure between hatches. In (*d*) it is stated that the detailed stress distribution at hatch corners is affected by the radius of curvature; this has been investigated experimentally by Røren (see Abstract No. 25,665, Sept. 1967).

It is concluded that a practical method of calculating torsional deformations and stresses has been devised.

26,531 **Small-Scale Grillage Tests.** CLARKSON, J. *R.I.N.A., Paper No. W9* (1966), *issued for written discussion* [9 pp., 8 ref., 4 tab., 4 graphs, 10 diag., 2 phot.]

The tests described were carried out on two ship-structural grillage models (both flat and singly plated, with T-bar stiffeners), made to about one-third or one-quarter scale in high-yield (Q.T.28) steel welded by the CO_2 shielded dip-transfer process. This process gives very low welding distortions and therefore permits the fabrication of relatively small models suitable for testing into the plastic range. The steel thicknesses ranged from 0·15 to 0·27 in. The models incorporated fillet welds, and one had butt welds in the plating; details at the intersections (which were of double-lug and bracket type) were correctly scaled. The scantlings, weld edge-preparations, and loading positions are shown in diagrams. The instrumentation for measuring stresses and deflections, and the test procedures, are described. The models were loaded elastically over the stiffeners, and into the elasto-plastic range over individual plate panels.

Readings of permanent set were obtained. Finally, they were loaded to plastic collapse. The grillages were supported at the corners only, and the loads were of a concentrated character.

The results are presented mainly in graphical form, and are compared with those of theoretical calculations assuming either half or all of the plating to be effective as flanges and taking account of shear deflections. It was found that the models could be loaded to beyond their theoretical plastic-collapse loads without any sign of fracture. The presence of butt welds in the plating had no noticeable effect on panel behaviour under a localised loading or on the overall behaviour up to the maximum loading applied. Within the elastic range, the results for loadings on the beam structure confirmed the earlier conclusion that calculations with " half-plate " effective breadth are sufficiently close for design purposes. With loading over the centre of a plate panel, some very high local bending stresses in stiffeners were observed; these were attributed to very small effective breadths (10–25% of the stiffener spacing) around the loaded panel. Data on the local plate-panel behaviour were also obtained. The plastic collapse of the grillages gave results which generally confirmed earlier work.

26,532 **An Approximate Stress-Calculation Method for Transverse Beams of Symmetrical Cross-Section, having Lightening Holes with Rounded Ends** (in German). WIEBECK, E., and HÄNERT, M. *Schiffbautechnik*, **17** (1967), p. 326 (June) [4 pp., 8 ref., 6 diag., 2 phot.]

SHIPBUILDING (GENERAL)

(*See also Abstracts No.* 26,568 *and* 26,569)

26,533 **A Comment on Low and High Freeboard Tankers.** BONEBAKKER, J. W. *Shipbuild. Shipp. Rec.*, **110** (1967), p. 475 (5 Oct.) [1 p., 1 ref., 2 tab., 1 graph]

The increase in tanker size has been accompanied by an increase in deadweight/displacement ratio, calling for relatively greater cubic capacity. This has led to many tankers being given relatively greater depth and freeboard, the latter being about 0·33 D (as for dry-cargo ships) instead of the 0·25 D formerly usual for tankers up to 30,000 tons d.w. The following data are tabulated for 29 recent tankers of 90,000–202,000 tons d.w.: deadweight, draught (d), L/B, L/D, and d/D. Seventeen of them have " low " and twelve " high " freeboard. When the values of d/D are plotted against L/D, they fall about a mean line whose equation is $d/D = 0{\cdot}03426\ L/D + 0{\cdot}28$. The range of L/B is 6·25–6·65 for the low-freeboard group (apart from three ships with higher values), and 5·90–6·25 for that with high freeboard. For the majority of ships in each group, the ranges are much narrower (6·40–6·50 and 6·00–6·10 respectively).

26,534 **A Diagrammatic Method for Determining Freeboard in Accordance with the 1966 International Convention on Load Lines** (in German). DIEKOW, A. *Schiffbautechnik*, **17** (1967), p. 422 (Aug.) [6 pp., 2 ref., 2 tab., 12 graphs]

In 1955, Danckwardt published a simple graphical method enabling freeboard to be determined in the initial design stage (see Abstract No.

12,077, Oct. 1956). The present Author discusses the revised freeboard requirements contained in the 1966 International Convention on Load Lines (see also Abstract No. 26,325, Apr. 1968), and presents a revision of Danckwardt's method (including new diagrams) designed to meet the new requirements. The method is fully explained, with an example.

26,535 **Calculation, to a First Approximation, of the Steel Weight of Tankers** (in Russian). KONTOROVICH, B. M. *Sudostroenie*, No. 6 (1967), p. 8 (June) [5 pp., 9 ref., 6 tab.]

Existing formulae and methods for estimating the steel weight of tankers in relation to basic data such as the main dimensions do not permit a study of the influence of the mechanical properties of the hull steel. Now that higher-strength low-alloy steels are commonly used, this is an important consideration.

In deriving a suitable formula, steel weight is here considered as the sum of the following three terms: G_1, the weight of the members providing overall strength (essentially bottom and deck plating and longitudinals—the influence of the sides and longitudinal bulkheads is represented by a coefficient); G_2, the weight of transverse members, stem and stern, longitudinal and transverse bulkheads, etc.; G_3, the weight of superstructures, deckhouses, and other metal structures above the upper deck. Formulae are derived separately for each of these. The final formula for steel weight (G) involves certain constants whose values have been ascertained empirically, the yield stress of the basic hull steel, the main ship dimensions and block coefficient, the dimensions or volumes of the superstructures, etc., and the section modulus of the upper deck. This last item can be found by calculation (if a structural midship-section is available), or from Classification Society Rules.

In order to check the validity and accuracy of the formula, a tabular comparison is made of steel-weight values for 17 tankers, (*a*) as calculated by the formula and (*b*) as obtained by adjusting their actual weights to eliminate the effects of special design features such as corrugated bulkheads and ice strengthening. The steel weights concerned range, roughly, from 3,000 to 25,000 tons. The table shows that the formula is satisfactory for first estimates (errors do not exceed 6%), and for analysis of the influence of changes in the main dimensions on steel weight. Another table shows the relative steel-weight reductions (estimated by the formula) for three of the tankers considered, if higher-strength steels of various yield points had been used instead of mild steel; they agree well with the results of a different calculation method.

It also appears that G_1 accounts for about 25%, and G_2 for about 68%, of total steel weight. Research into means of weight reduction has hitherto been concentrated on those structural items included in G_1; reduction of G_2 calls for more accurate methods of calculating transverse strength. Other figures given show that the use of low-alloy steel can reduce steel weight by as much as 17%, depending on the grade. It is also noted that G_3 (superstructures, etc.) is not a negligible item; in tankers of average size it is comparable with G_1. There is thus a need, especially in the smaller ships, to reduce superstructure weight, e.g. by the use of light alloys or plastics.

A table relating to four medium-size tanker designs (700–750 ft b.p., 0·81 C_b, normal ship steel) illustrates the effects of changes in main

dimensions and their ratios, and in the section modulus of the deck, on G_1, G_2, and G_3, and on their total. Reduction in L/B produces a substantial reduction in hull weight, and can be economically justified even though it adversely affects other ship characteristics. As expected, an increase in draught produces the largest reduction in hull weight. It has been suggested (see Abstract No. 20,793, Dec. 1963) that an increase in depth, making it possible to provide separate tanks for all ballast, leads to a reduction in design bending moment and increases the depth of the equivalent girder, so that there is no net increase in hull weight. Without denying the desirability, in principle, of building tankers with excess freeboard, the present Author points out that an increase in depth does in practice lead to an increase in steel weight (diminution of G_1 and G_3 being outweighed by increase of G_2), if the section modulus of the deck remains constant. Classification Society requirements make it impossible to reduce the section modulus of the equivalent girder. Even with minimum freeboard, the still-water bending moments of large tankers can be reduced by proper distribution of surplus cubic capacity.

See also Abstract No. 24,308 (June 1966).

26,536 **A Study of the Suitability of the Catamaran Form of Construction for Small Passenger-Ships and for Tugs** (in German). SCHIMKE, A., and KUSSEROW, P. *Schiffbautechnik*, **17** (1967), p. 305 (June) [3 pp., 5 ref., 2 tab., 6 diag.]

The Institut für Entwerfen von Schiffen, of Rostock University, has previously carried out investigations on the practicability of catamaran fishing-vessels (see Abstract No. 24,252, May 1966). The present article is a summary of a thesis submitted to the Institut in 1966, and is concerned with the suitability of the catamaran concept for small coastal passenger-ships and for tugs.

A tabular comparison is made (and discussed) between the characteristics of an existing single-hull passenger ship of length b.p. 39·2 m (129 ft), breadth 7·6 m (24·9 ft), and 398 metric tons displacement, and those of a catamaran design for the same service. The catamaran has a length b.p. of 30 m (98 ft), a breadth of 13 m (42·7 ft), and a displacement of only 259 metric tons; it can carry 453 passengers against the 305 of the single-hull ship (or 341 passengers against 230 for the longer voyages), but nevertheless affords more space per person for the passengers and crew of 21. The speed is 11 knots in each case, but the catamaran requires only 2 × 265 h.p. (metric) against the 2 × 300 of the conventional ship. Other advantages of the catamaran are a small reduction in design draught, and better stability and manœuvrability.

A similar comparison is made between an existing conventional harbour-tug of length b.p. 23 m (75 ft), breadth 7·6 m (24·9 ft), 239 metric tons displacement, and 920 metric h.p., and a catamaran tug design (length b.p. 19 m, i.e. 62 ft, breadth 11 m, i.e. 36·1 ft, 241 metric tons displacement, 2 × 400 metric h.p.). The catamaran has much the better stability, a particularly important characteristic for tugs, and offers more deck room and more accommodation space. With about 13% less power, the catamaran has about the same bollard pull (just over 10 tons) and free-running speed (10·8 knots approximately) as the conventional tug; this is due to the catamaran's higher propeller efficiency. The catamaran would also be more manœuvrable.

The article includes tabulated dimensional and performance data

additional to those mentioned above, and general-arrangement drawings are given of the two single-hull vessels and of the two suggested catamaran designs.

26,537 **Construction of N- and M-Series Vessels for Shell Tankers on the Slipways at N.D.S.M., Amsterdam.** *Motor Ship*, **48** (1968), p. 555 (Mar.) [5 pp., 1 tab., 7 diag., 8 phot.]

The 109,710-ton d.w. *Neverita*, the largest ship built to date in Holland, has been delivered to the Royal Dutch Shell group by the Netherlands Dock and Shipbuilding Co. (N.D.S.M.), Amsterdam. She is the 47th Shell tanker to be built by N.D.S.M. Her principal particulars are:—

Length, o.a.	265 m (869·4 ft)
b.p.	253 m (830 ft)
Breadth, moulded	40 m (131·2 ft)
Depth, moulded	19·62 m (64·3 ft)
Summer draught, underside of keel .	14·932 m (49 ft)
Corresponding deadweight . . .	109,710 tons
Cargo-tank capacity	4,734,888 cu ft
Clean-ballast capacity	682,245 cu ft
Propulsion machinery	Sulzer Diesel 9RD90
Output	18,000 b.h.p. at 118 r.p.m.
Cargo pumps, turbine-driven . .	2 × 4,500 tons/hr
Ballast pump, turbine-driven . .	1 × 2,200 tons/hr
Trial speed on summer draught . .	14·55 knots

The *Neverita* is generally similar to the 115,250-ton d.w. *Narica*, the first of the N-class tankers, which was built by the Swan Hunter Group and delivered to Deutsche Shell in Sept. 1967 (see Abstract No. 26,484, June 1968). The *Neverita*, however, has about 6 ft less beam and is more extensively automated than the *Narica*. This latter feature has permitted the issue of the first Lloyd's Register certificate for an unmanned engine room. The owners particularly required extension of the automation to guard against the risk of an electrical " black-out " in the event of the steam flow to the turbo-generator, or the electrical frequency, falling below a pre-set level. The main and auxiliary machinery is almost entirely automated, enabling the entire engine-room staff to be employed on routine day-work maintenance duties. The ship is also the first in the Shell fleet to have centralised control of the pump-room and cargo valves.

Construction complies with the requirements of Lloyd's Register Class ✠ 100 A1, and higher-tensile steel has been extensively used in the bottom-shell and deck plating. Scantlings in the tanks have been reduced by adopting an approved system of corrosion control. Shell Epicote paints have been applied in the tanks under controlled conditions of temperature and humidity, and also to the external shell.

There are five centre cargo tanks and ten wing tanks; No. 3 wing tanks are reserved for water ballast. Clean-ballast capacity is 18% of the total deadweight. A special slop-tank, for use in the ship's " load-on-top " system, is fitted in No. 5 centre tank. No cargo-heating coils are fitted.

The article includes photographs of the ship, the wheelhouse control console, the control-room console, the main engine and engine room, and the Babcock & Wilcox auxiliary oil-fired boiler The machinery control

room overlooks the engine room, and is on the same level as the engineer officers' accommodation.

The Netherlands Dock and Shipbuilding Co. are now engaged on the construction of four 210,000-ton d.w. tankers, three for Shell (M-class), and one for A. P. Møller, of Denmark. (See also Abstracts No. 26,436, May 1968, and 26,483, June 1968.) The principal particulars of the Shell ships, the first of which will be named *Melania*, are:—

Length, o.a.	1,067·3 ft
b.p.	1,017 ft
Breadth, moulded	154·75 ft
Depth, moulded	80·4 ft
Summer draught	62·3 ft
Corresponding deadweight	210,000 tons
Cargo-tank capacity	8,772,170 cu ft
Designed speed on summer draught	15·4 knots
Crew	45

Propulsion is by an Atlantique/Stal-Laval steam-turbine set of 28,000 metric s.h.p. at 86 propeller r.p.m. (see Abstract No. 26,436). The main boiler is an N.D.S.M.-built Babcock & Wilcox unit with a maximum evaporation of 100 tons/hr, and a normal continuous evaporation of 85 tons/hr. Superheater outlet conditions are 63 kg/sq cm (896 lb/sq in), 515° C (959° F). Electrical power is supplied at 440 V by a turbine-driven and a Diesel-driven alternator, both of 750 kW capacity. There is a 100-kW emergency Diesel set. There are four turbine-driven cargo pumps, each of 3,500 cu m (113,600 cu ft)/hr at 125 m (410 ft) head, one turbine-driven ballast pump of 4,250 cu m (150,090 cu ft)/hr at 30–35 m (98–115 ft) head, and two stripping pumps, each of 350 cu m (12,360 cu ft)/hr.

Each of the four 210,000-ton d.w. ships will be built in two parts, which will be launched separately (the forward part bow first) and joined when afloat. The after part will be launched first to enable machinery installation and fitting out to proceed while the less complex fore part is being built. The procedure for joining the parts is described, and the various stages involved are shown in diagrams. After careful ballasting to ensure correct trim and alignment, the two parts are drawn together and the butting surfaces enveloped on the outside of the shell by a U-section steel caisson. The caisson has watertight seals and can be pumped dry to allow welding to proceed from the inside of the shell. A similar method of afloat joining has been adopted by Mitsubishi (see Abstract No. 26,167, Feb. 1968).

The advantages of the two-part launching and afloat-joining method over more conventional methods of constructing large ships are stated and discussed. They may be summarised as (*a*) simplification of launching calculations and reduction in the risk of structural damage during launching (the launching trim of the separate parts can be adjusted to suit declivity of slipway), (*b*) more continuous employment of labour, and (*c*) less storage space and cranage needed, adjacent to the berth, for completed sub-assemblies. As fitting-out the after portion takes much longer than building the forward portion, and both should be ready at the same time, a relatively modest degree of prefabrication, using panel-type sub-assemblies not exceeding 150 tons each, was considered most

appropriate. The berth has therefore been equipped with a 150-ton portal crane with outriggers to cover the storage areas on each side. The berth and crane, with a ship under construction, are shown in photographs.

In addition to shipbuilding, N.D.S.M. have diversified their activities to embrace the construction of offshore drilling rigs, and of towers, heat exchangers, pressure vessels, gate valves, etc., for the oil, chemical, and petro-chemical industries. The firm can undertake the installation of complete plants for these industries.

26,538 ***Columbialand*—Prototype Packaged-Lumber Carrier.** *Shipp. World & Shipb.*, **161** (1968), p. 73 (Jan.) [7 pp., 3 tab., 11 diag., 6 phot.] See also *Motor Ship*, **48** (1968), p. 560 (Mar.) [4 pp., 5 tab., 5 diag., 4 phot.]

The 24,900-ton d.w. *Columbialand* is the first of five sister ships to be completed for Scanscot Freighters, an international service pool formed by several Scandinavian and Scottish owners whose names are given. The ship has been handed over by the builders, Öresundsvarvet AB, Landskrona, Sweden, to AB August Leffler, Gothenburg, who will operate the pool. Three similar ships are being built by this firm, and a fifth by Charles Connell & Co. (Shipbuilders) Ltd, Glasgow. All the Swedish-built ships will have Götaverken main engines and the British ship a Sulzer engine.

The *Columbialand* is an all-aft single-deck bulk-cargo ship intended primarily for the carriage of packaged timber, although other cargoes such as grain, phosphates, and standard 20 or 40 ft containers can also be accommodated. Packaged timber can be stowed on deck up to a height of 23 ft.

The ship has a soft-nosed clipper stem with a conventional forefoot, a short raised forecastle, a poop, and a wide V-section transom. Principal particulars are:—

Length, o.a.	574·8 ft
b.p.	550 ft
Breadth, moulded	87 ft
extreme	87·4 ft
Depth, moulded	45 ft
Draught, summer	31·1 ft
Deadweight, corresponding	24,900 tons
Draught, summer, timber freeboard	32·3 ft
Deadweight, corresponding	26,335 tons
Gross tonnage	17,437
Block coefficient	0·78
Cargo capacity, grain	1,226,550 cu ft
bale	1,198,795 cu ft
Heavy fuel oil	1,321 tons
Diesel oil	322 tons
Machinery output	11,400 b.h.p. at 124 r.p.m.
Speed	15·5 knots
Classification	Det Norske Veritas ✠ 1A1 "T" "F" "Ice C"

When the 1966 International Load Line Convention rules become

operative, the deadweight on summer freeboard at 32 ft 5 in draught is expected to be 26,400 tons, and that on timber freeboard at 33 ft 6 in draught about 27,800 tons.

There are six cargo holds, each with two large side-by-side hatches which allow practically the entire deck area above the hold to be opened up. Details are shown in a general-arrangement drawing. Because of the exceptional size of the hatch openings the hatches, deck plating, and deck longitudinals have been suitably strengthened and the hatch-coaming corners are of the Kennedy cast-steel type (see also Abstract No. 23,856, Jan. 1966). An 8·25-ft deep centre-line deck girder eliminates the need for grain feeders. Corrugated bulkheads separate the holds, and No. 3 has an additional portable transverse bulkhead for the separation of different cargoes. The tank top throughout the hold space has been stiffened for grab-discharge of ore, etc. Nos 2 and 6 holds have upper wing tanks for water ballast, and Nos 3, 4, and 5 have deep wing tanks. Heavy oil is carried in No. 1 (raised) double bottom and in wing tanks in way of the engine room.

All hatch covers are of ASCA (Associated Cargo Gear AB) construction. They are hydraulically operated and have hinged longitudinally-folding panels to suit a variety of loading or discharging conditions. Complete opening of all hatches can be effected in six minutes. AB Regnsegel portable rain-tents (shown in a diagram in *Shipp. World*) are carried in the ship and can be positioned over any of the hatches to give weather protection during cargo handling.

There are five Hägglund hydraulically-operated tower-mounted deck cranes, positioned between the holds; each has a capacity of 10 tons at 72·2 ft radius. Special equipment for turning a suspended load to the desired position for stowing is fitted to each crane hook; it is powered by a 1-h.p. electric motor and gearing, remotely controlled by the crane operator. There are also two 10-ton derricks on the fore-side of the superstructure and an 8-ton hydraulic derrick crane on the forecastle.

The air-conditioned accommodation is to a high standard; some particulars are given in *Shipp. World*, which also gives details of the navigation equipment.

The main engine is a six-cylinder Götaverken 750/1600 VGS-6U Diesel with an output of 11,400 b.h.p at 124 r.p.m. It has two Brown Boveri turbochargers and will run on heavy fuel of up to 3,500 sec Redwood No. 1 at 100° F. The installation complies with Class EO of Det Norske Veritas for a periodically unmanned engine room. An 18-ft diameter KaMeWa controllable-pitch propeller (in Novoston) is fitted. In order to obtain, when going astern, the same direction of rotation as for a fixed-pitch propeller, the main engine has been arranged to turn anti-clockwise instead of clockwise as usual.

There is a Sanea exhaust-gas boiler and an Aalborg oil-fired boiler; their steam capacities and pressures are stated. Capacities of the two Götaverken air compressors and of the numerous pumps, etc., are given in *Shipp. World.*

Electrical power is provided by three ASEA 488-kVA, 450-V alternators, each driven at 600 r.p.m. by a Bergen 570-b.h.p. Diesel engine; these are mounted aft of the main engine at top-platform level. There is also a 93-kVA emergency alternator driven by a Volvo Penta Diesel engine.

The main engine and the c.p. propeller can be controlled either from

the bridge or from a machinery control room on the starboard side at top-platform level.

Both articles include general-arrangement drawings, a deadweight scale, hold capacity tables, etc. *Shipp. World* also gives a shell expansion drawing and an engine-room layout.

26,539 **"Liberty" Ship Replacements: the Dutch 14,000-ton d.w. "Unity" Vessel Proves Successful.** *Motor Ship*, **48** (1968), p. 567 (Mar.) [2 pp., 2 tab., 1 diag.]

Jadranska Slobodna Plovidba, of Split, Yugoslavia, have placed an order worth about £7·5 million with Cockerill Y^rds Hob_ken S.A., Antwerp, for six "Unity" class Liberty replacem· · ships. The design was developed by Sea Transport Engineering : '., marine consultants of Amsterdam, on behalf of Gebr. Stork N.V., Hengelo, Holland. The latter firm required a design with very low operating costs and utilising as far as possible products of the VMF/Stork-Werkspoor Group. The standard hull and machinery specification is set out in tabular form, and general-arrangement drawings are given. The principal particulars are:—

Length, o.a.	137·8 m (452·2 ft)
b.p.	132 m (433 ft)
Breadth, moulded	20·5 m (67·25 ft)
Depth, to main deck	12·6 m (41·3 ft)
to second deck	9·3 m (30·5 ft)
Draught	8·82 m (28·9 ft)
Tonnage, gross	8,800
Cargo capacity, bale	700,000 cu ft
grain	755,000 cu ft
Homogeneous stowage rate	54 cu ft/ton
Bunker capacity	1,325 tons
Fuel consumption, all purposes	21 tons/day
Service speed	14 knots

The engine room and superstructure are located three-quarters aft as in the Austin and Pickersgill SD 14 (see Abstract No. 24,599, Sept. 1966). This arrangement possesses important advantages which outweigh disadvantages compared with the r ɔre commonly adopted all-aft layout. These advantages and disadvantages are stated and discussed: *inter alia*, the engine room can be shorter, less piping is required, and trimming in various load conditions is greatly simplified. The ship has a straight raked stem, a conventional forefoot, a closed stern frame, and a wide, flat, vertical transom. There is no forecastle and no poop. The four holds, three forward and one aft of the machinery space, are all of about the same capacity. All the cargo hatches on both decks have the same dimensions (15·25 8 m, i.e. 50 · 26·25 ft); the covers on the upper deck are of MacGregor single-pull type, and those on the second deck are wooden with steel shifting beams. Each hold is served by four 3/5-ton derricks with electric winches. Standard containers can be carried in the tweendecks, and liquid cargoes in deep tanks in No. 3 hold.

Adequate stability can be maintained, and change of trim due to fuel consumption need not be corrected by ballasting. Calculated trim and GM for loaded and ballast conditions are given in a table.

Propulsion is by a Stork SW5 · 63/135 Diesel engine with an output of

5,500 b.h.p. at 140 r.p.m. It is bridge-controlled through a pneumatic system, and stops automatically when lubricating-oil pressure drops below an accepted level. Electrical power is supplied by three 220-kVA, 450-V Diesel-alternator sets with Stork RO 156 k engines (1,200 r.p.m.). There is a Spanner exhaust-gas boiler and an oil-fired boiler. The salt-water and fresh-water cooling circuits, and the lubricating-oil circuits, of the main and auxiliary engines are controlled thermostatically. Audible and visual alarms are provided for the lubricating-oil, cooling-oil, and cooling-water temperatures of the main engine.

The "Unity" class ships can be operated by a crew of 24 (which includes a pilot), but there is accommodation for 28 in the flat-walled five-tier superstructure. Messing is arranged for self-service, requiring a catering staff of only three men. The manning scale recommended is shown in the specification.

26,540 ***Dragon* [Passenger and Car Ferry for Le Havre—Southampton Service].** *Shipbuild. Shipp. Rec.*, **110** (1967), p. 79 (20 July) [5 pp., 2 tab., 3 diag., 11 phot.]

The *Dragon* is the first of two car and passenger ferries for the Southampton—Le Havre service known as Normandy Ferries; her operators are Southern Ferries Ltd, of Southampton. Work on the second ship, the *Léopard*, started in June 1967, the month in which the *Dragon* entered service. The *Léopard* is being operated by Société Anonyme de Gérance et d'Armement, Paris. Both vessels were built by Ateliers et Chantiers de Nantes and fitted out by Dubigeon-Normandie. *The Dragon's* principal particulars are:—

Length, o.a.	134·1 m (440 ft)
b.p.	121·5 m (398·6 ft)
Breadth, moulded	21·24 m (69·7 ft)
Depth, moulded, to upper deck	11·75 m (38·5 ft)
to main deck	6·7 m (22 ft)
Draught	4·75 m (15·6 ft)
Deadweight	2,387 tons
Propulsion power	2 × 4,440 b.h.p. at 370 r.p.m.
Service speed	15 knots
Classification	Lloyd's ✠ 100 A1 ✠ LMC

The *Dragon* has a soft-nosed and raked stem, a transom stern, and three superstructure decks above the flush upper deck. The car deck (main deck) has an uninterrupted run to the collision bulkhead. It is free of obstructions except for a narrow centre-line casing, and has two stowable car decks in the wings whose sections can be hinged up hydraulically when not required. The maximum numbers of passengers, cars, and trailers that can be carried are respectively 850, 360, and 60. Vehicles enter or leave the car deck via hydraulically-operated stern doors and a stern ramp; above this is a similar hinged ramp for the upper deck. Special emphasis has been given to the carriage of freight on large trailers. A fleet of ships' trailers is available, to which forwarders or hauliers can transfer their own containers or pallets, etc., at the quayside where facilities are available for lifting loads up to 29 tons. Vehicles containing dangerous cargoes can be carried on the upper deck aft where space is available

for 15 trailers, and where also vehicles carrying outsize loads up to 22 ft wide and 21 ft high can be accommodated. The deck has a large flush hatch in this region. The linkspan to the shore is connected as required to the upper or lower stern ramps; it is wide enough (22 ft) to maintain two-way traffic, and loads up to 205 tons (not exceeding 13 tons per axle) can be accepted. A passive roll-stabilising tank system is installed forward of the engine room (which is amidships).

The welded hull is transversely framed, and in the garage and engine-room web frames are provided at each sixth frame space. The scheduled service will include night crossings; sleeping accommodation is provided for 180 passengers in four-berth sleeperettes, and for a further 96 in two- and four-berth cabins below the car deck. Passenger amenities include air-conditioning of all living spaces, a nursery, duty-free shops, ship-to-shore telephone, and television equipment designed for the reception of both English and French programmes and for use in conjunction with automated teleciné equipment to show films and slides in the various passenger lounges.

Propulsion is by two 12-cylinder SEMT-Pielstick PC2V turbocharged Diesels, each with a maximum rating of 5,580 b.h.p. at 500 r.p.m. and coupled directly to a KaMeWa controllable-pitch propeller. The engines and propellers, the bow-thrust unit, and the twin rudders can be controlled from either bridge wing as well as from the wheelhouse. The four constant-tension mooring winches can also be remotely controlled from the bridge wings.

Electric power at 380 V is supplied by three Duvant sets, each comprising a 480-kW/600-kVA alternator driven by an 800 h.p. turbocharged Diesel running at 750 r.p.m.

26,541 ***Bore VI.* Multi-Purpose Cargo Vessel Built in Finland.** *Shipp. World & Shipb.*, **160** (1967), p. 2083 (Dec.) [3 pp., 1 tab., 4 diag., 2 phot.] See also *Shipbuild. International*, **10** (1968), p. 4 (Jan.) [1½ pp., 1 tab., 2 diag., 2 phot.]

The *Bore VI* is a versatile small cargo-vessel, with the emphasis on roll-on/roll-off facilities; she was built by Rauma-Repola Oy, of Rauma, Finland, for the Bore Steamship Co., of Turku (Åbo). She is intended for liner service in Baltic and Scandinavian waters, and for the Atlantic Container Line's feeder route between South Finland and Gothenburg. She is all-welded, meets the requirements of Lloyd's Class ✠100 A1, ✠LMC, Ice Class 1, and complies with the 1960 Safety Convention. The structure is based on a scantling draught of 6·2 m (20·3 ft), which would bring the sill of the stern door to water level. The principal particulars are:—

Length, o.a.	88·2 m (289·4 ft)
b.p. (on design w.l.) .	80 m (262·5 ft)
Breadth, moulded	14·5 m (47·6 ft)
Depth, to upper deck . . .	8 m (28·9 ft)
to tween deck . . .	5·6 m (18·4 ft)
Draught, to design w.l. . .	5·2 m (17·1 ft)
Corresponding displacement .	3,870 tons
Draught, to tonnage mark . .	5·556 m (18·2 ft)
Corresponding deadweight . .	2,350 tons

Maximum deadweight	3,000 tons
Tonnage, gross	2,753/1,473
net	1,555/661
Cargo capacity, bale . . .	171,500 cu ft
Service speed	14·5 knots

The hull has a raked stem with rounded forefoot. There is a forecastle, and a quasi-poop deck (partly cut away right aft) with superstructure above. These are joined by continuous raised hatch coamings forming a trunk; this gives the inboard lanes of the tweendeck a headroom of 4·35 m (14·3 ft). The stern aperture, 4·42 m (14·5 ft) square, is provided with a double-flap ramp/door. The drive-in passage, with stores and machinery casings on either side, has minimum clearances of 4·1 m (13·4 ft) in height and 4·4 m (14·4 ft) in width; container-carrying vehicles can therefore be driven aboard. Below it is the engine room. Forward of this region, the tweendeck is clear to the collision bulkhead and the lower hold to the ballast deep tank, apart from two side-by-side lift shafts linking these two levels at about L/3 from the bow; the lifts and their flap-doors are equipped with automatic roller conveyors. These lifts are intended mainly for handling forest products; the ship can carry about 2,000 tons of paper or 2,500 tons of pulp.

The mechanised steel natch covers on the trunk and the tweendeck were supplied by Navire Oy, who also provided the two mechanised stowable car decks. One of these, in the lower hold, has sections which fold against the ship's sides; the other (at upper-deck level, inside the trunk) stows in the forecastle. All vehicle levels are accessible by ramps. A cargo of 300 medium-sized cars can be carried. The cargo spaces can be ventilated by four reversible axial-flow fans at 15 air changes/hr. A 5-ton ASEA deck crane is mounted on the forecastle, another above the lifts, and one of 10 tons just forward of the bridge front. The pole-changing electric windlass, and the two self-tensioning capstans aft (these have Ward-Leonard drives) were made by the shipbuilders. The complement of 17 all have single cabins; heating of the ventilation air is supplemented by hot-water radiators.

The propulsion machinery consists of two six-cylinder unidirectional MWM TbRHS 345S engines, each rated for 1,650 b.h.p. at 500 r.p.m. They are equipped to burn heavy oil of up to 400 sec Redwood, and drive the single Escher-Wyss controllable-pitch propeller at 200 r.p.m. through Pneumaflex KAE 260 W pneumatic clutch/couplings (see Abstract No. 25,928, Nov. 1967) and a Lohmann & Stolterfoht gearbox. The engines are started locally, but engine speed and propeller pitch are subsequently controlled from either the bridge or the machinery control room through a pneumatic system; the manœuvres are recorded by an automatic printer. Safety provisions include automatic clutch disengagement on exceeding a pre-set torque, and emergency engine-stop facilities in the wheelhouse. There are three 205-kVA Diesel-alternator sets, one of which is pre-set to start on loss of mains voltage. The cargo spaces and engine room have CO_2 fire-extinguishing systems, and appropriate fire detectors and alarms.

The *Shipp. World & Shipb.* article has the more extensive set of general-arrangement drawings; *Shipbuild. International* includes a list of equipment suppliers.

26,542 ***Tyro*—Versatile Dutch Cargo Vessel for the Short Sea Trades.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 475 (5 Apr.) [5 pp., 1 tab., 4 diag., 7 phot.]

The motor ship *Tyro*, completed at the end of 1967, was built by Van der Giessen—de Nord (of Krimpen a.d. Ijssel, Holland) for Maatschappij Zeevaart (of Rotterdam). She was built to the owners' design for their service between Rotterdam, Le Havre and Dublin, and is specially arranged for pallet, container, and refrigerated cargoes and for carrying cattle. Her principal particulars are:—

Length, o.a. . . .	84·23 m (276·3 ft)
b.p. . . .	77 m (252·6 ft)
Breadth, moulded .	14·3 m (46·9 ft)
Depth, moulded .	9·68 m (31·8 ft)
Draught . . .	4·5 m (14·8 ft)
Deadweight . .	1,500 tons
Bale capacity . .	5,090 cu m (179,750 cu ft)
Speed	14 knots
Classification . .	Bureau Veritas ✠ 1 3/3 L.I.I., A & CP

The *Tyro*, of all-aft layout and nominally a single-decker, has two higher decks, the upper one terminating just short of the transom stern. Below the main deck (which forms the lower tweendeck), four transverse bulkheads divide the hull into the following main compartments: fore peak, bow-thruster compartment, lower hold, engine room, and after peak; in addition, there is a dry tank at the stern. There is no sheer or camber, and all hatch covers are flush-fitting except those on the upper deck. The cargo spaces are clear except for centreline pillars and certain casings aft. A refrigerated compartment, at the forward end of the upper tweendeck, has a capacity of 400 cu m (14,125 cu ft), and an operating temperature between $-10°$ and 20° C (14° and 68° F). The Grenco direct-expansion refrigeration system has three fully-automatic Grasso R-22 compressors. The cargo spaces have mechanical ventilation. Pens can be erected in the lower hold for 300 head of cattle.

There are three hatchways, including one for the refrigerated compartment. The hatch covers, all hydraulically operated, are of steel; the end-rolling type is used on the upper deck and hinged panels elsewhere. On the starboard side, there is a bottom-hinged cargo side-door between Nos 2 and 3 hatches and another aft of No. 3. These hydraulically-operated watertight doors, which are 4·7 m wide by 5·8 m deep (15·4 by 19 ft), can be lowered to form 10-ton ramps between ship and shore; the hinge level can be adjusted vertically to suit the quay height. Inboard of each side-door are side hatches with hinged steel covers, in the upper deck and in the upper tweendeck; these permit truck-to-truck operations if the quay height prevents fork-lift trucks from driving directly on and off. There is a 5-ton hydraulic deck-crane between Nos 1 and 2 hatches and a 15-ton one amidships, i.e. forward of No. 3 hatch. When the 15-ton crane is trained outboard, list is counteracted by ballasting (there are special wing tanks in way of the lower hold). The ship sides are protected by a steel fender which has upward extensions to protect the cargo-doors.

The wheelhouse, chartroom, and radio room are combined; the wheelhouse instruments and controls (which include those for the controllable-pitch propeller and the bow thruster, both of which can also be controlled

from the engine room) are all mounted c consoles. All crew members have single cabins. There is accommodation for 12 passengers. All accommodation is air-conditioned and is to a higher standard than is usual in the short sea trades. There are two 40-person g.r.p. lifeboats and a 20-person inflatable liferaft.

The Lips four-bladed c.p. propeller is driven by two MWM type Tb RHS 345S non-reversing six-cylinder four-stroke engines, through Pneumaflex KAE clutch/couplings and Lohmann & Stolterfoht 2 : 1 single-helical reduction gearing. The couplings protect the drive from torsional vibration and permit instant connection and disconnection. The engines run on Diesel oil. Each has a continuous rating of 1,500 h.p. at 495 r.p.m.; a 240-kW 380-V alternator is driven off the port engine. At sea, both engines normally provide the propulsion power, with the shaft generator feeding into the mains. In this mode, engine speed is automatically kept constant for all pitch settings. For river and similar passages, the port engine is disengaged from the propeller drive and the shaft generator feeds the bow-thruster motor. In this mode, engine speed and pitch are varied simultaneously by a combined control. The bow-thruster, a Tornado unit, is driven by a Heemaf motor and can provide a thrust of 2½ tons at 1,450 r.p.m.

There is a machinery control room in the engine room. The pneumatic/electric engine-speed/propeller-pitch control system is of Westinghouse design. Propeller pitch control from the control room is "time-dependent", i.e. when the desired pitch or engine load is reached the speed/pitch lever must be returned to neutral. The bridge speed/pitch control is "way-dependent", i.e. the pitch setting is regulated in proportion to the pneumatic signal from the control lever. Bridge control can be overridden by the engine room; however, when the engine-room control has been returned to neutral, thepitch corresponding to the bridge setting is automatically resumed. The engines are started locally.

In addition to the shaft alternator, there are two 152-kW 380-V Diesel sets and a 30-kW emergency set which starts automatically if the mains voltage drops below 80% of the normal. The article gives further information on the machinery installation and its equipment. Automatic control is provided for many of the pressures and temperatures. There is an extensive alarm system covering the main-engine and auxiliary-engine cooling and lubrication systems, reduction-gear lubrication, clutch/coupling and control-system air-pressures, various tank-levels (including that of the bow-thruster lubricating-oil tank), propeller hydraulic-oil pressure, pump operation, electricity supply, fuel-separator water-seal, engine-room bilges, and steering-gear oil supply. Other alarms include those for leakage at the cargo side-doors.

The article includes general-arrangement drawings of the ship, and drawings showing the engine-room layout.

26,543 **Short-Sea Roll-On/Roll-Off Ships of Advanced Design.** *Motor Ship*, **48** (1967), p. 319 (Oct.) [4 pp., 1 ref., 4 diag., 9 phot.] See also *Schiff u. Hafen*, **19** (1967), p. 536 (Aug.) [4 pp., 1 tab., 6 diag., 8 phot.]

The *Antares* has entered service between Bremen, Hamburg, and Hull. She was built by Schlichting Werft, Lübeck-Travemünde, and, with her sister ship the *Arcturus* (built by Büsumer Werft, a subsidiary of

Schlichting Werft) will be operated by her owners, Argo Reederei Richard Adler & Söhne, Bremen, in a joint service with Associated Humber Lines Ltd. The agents are able to accept cargoes for shipment up to noon on the day of sailing. The ships have been designed as 500-ton gross "paragraph" vessels with a high cubic capacity. Their principal particulars are:—

Length, o.a. . . .	76·42 m (250·75 ft)
b.p. . . .	68 m (223·1 ft)
Breadth, moulded . .	13 m (42·6 ft)
Depth, to shelter deck .	9·92 m (32·5 ft)
to main deck .	4·25 m (13·9 ft)
Summer draught . .	4·22 m (13·8 ft)
Corresponding deadweight	1,123 tons
Gross register . . .	499·87
Capacity, grain . . .	5,170 cu m (182,580 cu ft)
bale . . .	4,830 cu m (170,570 cu ft)
Service speed . . .	14·6 knots
Endurance	5,000 n. miles
Classification . . .	Germanischer Lloyd. GL ✠ 100 A4E 1 ✠ MCE 1

The *Antares* has a soft-nosed clipper stem, a raised forecastle, and a broad and flat vertical transom to accommodate the hydraulically-operated stern ramp/door. Containers and other cargo can be handled by quay cranes through two large (18 ft wide, 46 and 30·5 ft long) hatches with flush covers in the weather deck, and two smaller flush hatches in the main deck below. The hatch covers are of Greer hinged folding type. There are no pillars or stanchions in the cargo spaces. The main deck can sustain loads of 13 tons at 1·3 m (4·27 ft) axle centres, and fork-lift trucks of 5 tons axle load. The double bottom is 1·3 m deep. The ship's middle body is almost straight-sided, with a chine at about the load waterline.

Although most of the cargo will be palletised, a total of 70 standard 20-ft containers can be carried in the lower hold (one tier) and the tweendecks (two tiers). The stowage arrangements are shown in a sectional drawing. Cargo can be loaded into or discharged from the lower hold by the "truck-to-truck" method, through the main-deck hatches. Vehicles can reach the weather deck from the stern door via a ramp which is hinged at weather-deck level just forward of the superstructure; this ramp can be lowered between the port and starboard machinery casings. (There is a door in the superstructure front.) When not in use, the ramp is secured under the deckhead and forms a watertight seal for the deck opening.

The propulsion machinery is a 16-cylinder M.A.N. V8V 22/30 ATL Diesel engine, rated at 2,200 b.h.p. at 900 r.p.m. It drives a KaMeWa controllable-pitch propeller at 275 r.p.m. through a Renk reduction gearbox. The normal electrical load at sea is met by a Siemens 76-kVA alternator mounted on the gearbox and belt-driven from a gearbox layshaft. A 162-kVA Diesel set supplies additional power when the Tornado bow unit (of 1·7 tons thrust) is being used; it is arranged for automatic starting in the event of breakdown of the shaft-driven alternator. For emergency and harbour use there is an air-cooled 48-kVA Diesel set.

A portable cover in the main deck (16·4 ft × 8·3 ft) enables the main engine to be removed as a unit if necessary.

The Tornado lateral-thrust unit is operated from the bridge wings or from the main control console in the wheelhouse (which has a wide field of view). Control of the Brown Boveri steering system for the Atlas Telenaut steering gear, and of the propeller pitch, is also effected from this console, which also has an emergency-stop button for the main engine. The navigation and wireless equipment is described. The manning scale for the crew of 12 is given. All personnel have single-berth cabins.

The article includes general-arrangement drawings, and photographs of the ship's interior, hatch and ramp arrangements, and the bridge instrumentation. The *Schiff u. Hafen* article includes structural section drawings.

26,544 **Pushed Barges with Membrane Bottoms** (in German). POHLANDT, G. *Schiffbautechnik*, 17 (1967), p. 497 (Sept.) [1½ pp., 3 ref., 2 diag., 2 phot.]

Two prototype barges, differing in design but each having an unstiffened " membrane " bottom, are giving satisfactory service on the East German inland waterways. They were completed in 1966 for use in pushed barge trains. They are not intended to be used as leading barges, and for this reason, and for simplicity of construction, they are " brick-shape "; the turn of bilge is square. Like other barges in service in East Germany, their length and breadth are 32·5 and 8·2 m (107 and 27 ft); their loaded draught is 2 m (6·6 ft) at the side, and somewhat more along the centreline because of the sag of the membrane bottom. One prototype, the RSW, has a cargo capacity of 535 cu m (18,893 cu ft); that of the other, the FAS, is 614 cu m (21,683 cu ft).

Transverse " half-height " bulkheads divide the cargo space into " holds ". The sides of the holds in the RSW barge are vertical and flush with the coamings; outboard of the hold, on each side, is a box-structure longitudinal formed by the side of the hold, the side of the barge, the outboard part of the barge bottom, and the stringer plate. In the FAS barge, the holds extend to the barge sides; triangular longitudinals run along the top and bottom, the bottom one (which is much the larger) being formed by the lower part of the barge side, the outboard part of the barge bottom, and plating angled to give a hopper shape. This difference in construction results in the membrane bottom of the FAS barge being less highly loaded (and having less sag) for a given weight of cargo, and the steel weight of the barge is about 10% less.

A brief account is given of tests in which brown coal was tipped into the barges, and of strain-gauge and deflection measurements made in dry dock while one hold was being filled with water (up to a level representing almost a three-fold overload in the case of the FAS barge). The results of the measurements confirmed the validity of the assumptions made for the structural calculations.

There are not, as yet, any Rules or recognised calculating methods for this type of construction. The two barges have a DSRK experimental classification, but will, subject to their satisfactory service, have a formal classification later. Further barges with membrane bottoms are to be built.

26,545 **Dynaplane Design for Planing Craft.** *Shipbuild. Shipp. Rec.*, **110** (1967), p. 395 (21 Sept.) [1 p., 6 diag.]

An account is given of a new type of planing craft developed by the U.S. Naval Ship Research and Development Center (formerly the David Taylor Model Basin); it is known as the Dynaplane design. It has a shallow step amidships. The forward lifting surface is designed to carry 90% of the total weight; it has S-form longitudinal curvature based on theoretical analysis aimed at developing the required lift on a very small wetted area. The remaining 10% of the weight is carried by an adjustable planing surface (also called a stabiliser) at the stern; this device was invented by J. Plum. Its vertical position can be varied by means of a pneumatic piston inside the hull; as the stabiliser is moved downwards, its angle of attack automatically changes from negative to positive. At low speeds the stabiliser is held in a retracted position against the hull. When the craft is planing, the adjustment makes it possible to control trim angle; in fairly smooth water the trim angle for least drag would be chosen, whereas in rough water trim would be adapted to the wave condition and relative heading. In head seas a small trim angle minimises impact accelerations; in following seas a fairly large one is needed to prevent the bow burying itself in the waves.

Compared with a conventional planing craft, wetted area is reduced by over 75% and drag by 50%; this has been confirmed by model tests. The principle is applicable to a wide variety of naval, commercial, and pleasure craft; some of the possibilities (swamp boat, patrol boat, personnel transport, landing craft) are shown in sketches.

See also item C in Abstract No. 26,209 (Mar. 1968).

INDUSTRIAL AND ECONOMIC INFORMATION

26,546 **The Use of Simulation Models in the Investigation of Tramping.** BRAUER, G. *Shipbuild. Shipp. Rec.*, **111** (1968), p. 471 (5 Apr.), and p. 511 (12 Apr.) [6 pp., 15 graphs]

The shipowner and the shipbuilder both have an interest in obtaining reliable predictions on the economics of Liberty-ship replacements. In the case of the Blohm & Voss *Pioneer I* and *Pioneer IV* (13,600 and 22,000 tons d.w. respectively; see also Abstract No. 26,303, Apr. 1968), mathematical models were used to simulate tramping operations, and computer calculations were made on the economic feasibility of these ships for tramping under different operating conditions; some statistical techniques were employed.

The Author (of Blohm & Voss) discusses this problem of profitable operation, with particular reference to building costs. Results of the simulation study are presented in curves, and are explained in some detail.

The results indicate an unsatisfactory state of affairs. Even with series production, it is hardly possible for the builder to sell at a price satisfactory to both builder and buyer. The solution appears to lie in the production of longer series, but no yard will favour this course without guarantees covering an adequate length of series. In view of the close connection between the shipping and the shipbuilding aspects shown in the results of this study, it appears to be very advisable that representatives of the two industries should try to produce a solution acceptable to both; the

owners require a cheap ship and the builders require information as to the types of ship which can be built in a long series and sold at a suitable price. The study results may be considered as proving that financial assistance is required in order to ensure competitiveness in the international market.

A practical conclusion from this study is that, to avoid the excessive commercial risk of building long series, the ships should be diversified but based on components produced in long series. This is the aim of the Blohm & Voss Pioneer multi-carrier system. Though similar attempts in the past have miscarried, it is considered that world economic development has now reached an appropriate stage for this type of production to be successful.

26,547 **Improving the Prospects for United States Shipbuilding—Interim Report, November 1967.** *Publication issued by the Center for Maritime Studies, Webb Institute of Naval Architecture, New York* [24 pp., 1 ref., 1 graph, 3 diag., 10 phot.]

This publication sets out the interim findings and recommendations of a Study Group which is investigating the present state of the U.S. shipbuilding industry with a view to improving its prospects. The industry is severely handicapped by high labour costs, and also by the fact that a stable production situation has not been attained in peacetime shipbuilding. Since 1945 both merchant and naval construction programmes have fluctuated greatly, a situation which has not encouraged large capital outlay on new facilities and equipment. Little effort has been made to introduce series production of standardised ships or components. Productivity (added value per man-hour) is lower than in many other U.S. industries.

To effect a substantial improvement, intensive efforts will be required over a period of several years. The essential prerequisite is a stabilised long-term construction programme. Capital improvements designed to reduce costs should then be made. Careful study is needed to determine the nature and extent of investments that would be optimum under stable production conditions in the U.S. It is felt that flexible multi-purpose yards are more appropriate than highly-specialised Arendal-type assembly lines. The number of vessels of similar design to be built in each participating yard should be as large as practicable, utilising as far as possible joint designs for different owners.

" Design for production " is recommended; this requires time, at the beginning of every programme, for careful ship design and production planning, with associated simplification of structure and outfitting, modular construction, and standardisation of components. (One interesting possibility is the use of a single standard afterbody for a range of ship types and sizes.) There should be continuing improvement in cost accounting systems, production engineering, and production planning and control systems. Modern techniques (some of which are noted and illustrated) should be adopted in transport, fabrication, and assembly. Research and development programmes, aimed at improved ship design and building techniques for easier and more economical production, should be pursued. They require co-operation between industry, labour, and government in obtaining adoption of the new techniques developed.

How near U.S. shipbuilding can come, under favourable conditions, to being competitive with that of other countries is difficult to predict, but

an estimate will be made during this study. As wage levels are rising faster in other countries, the situation should improve somewhat as time goes on.

See also Abstract No. 26,492 (June 1968).

26,548 **Economics of Push Towing Conducted for the Yawata-Hikari Barge Line.** SAWA, M. *Soc. N.A.M.E.* (*Great Lakes and Great Rivers Section*), *paper presented* 26 Jan. 1967 [30 pp., 4 tab., 5 graphs, 8 diag., 4 phot.]

The Author gives a comprehensive description of the system adopted by the Yawata-Hikari Barge Line (Japan) for transporting about 600,000 tons of 9 or 12 m (29·5 or 39·4 ft) steel billets annually from Yawata (near Shimonoseki) to Hikari, a coastal voyage of 64 miles in dangerous waters. Waves up to 30 m (98·4 ft) in length and 3 m (9·8 ft) in height may be encountered. The system uses three barges of 2,000 tons d.w. and 223 ft length b.p.; these are moved (singly) by the pusher tug *Hakko Maru*, of 49 ft b.p., which is propelled by two geared Diesels, each of 650 b.h.p. at 320 r.p.m. The tug is of the " swallow-tail " type, in which the after part of the hull is of catamaran construction; two propellers are fitted, with Kort-nozzle rudders. The barges, which have hatches measuring 36·8 by 5·2 m (121 by 17 ft) and a large ballast capacity, operate on a 24-hour schedule, with one at each terminal and one in transit. The tug/barge connection is of a spring/hydraulic type developed by Mitsubishi (patent applied for); it is shown in diagrams Drawings, photographs, and technical details are given of the barges and pusher tug; also, the results of turning-circle and zig-zag trials of the tug/barge combination are presented graphically.

The paper contains detailed discussion of the economics of push towing in Japanese coastal shipping generally (comparisons are made with transport by coasters) and in this particular service. The considerations leading to the chosen craft particulars and operating schedule are explained (they include calculations of Required Freight Rate).

SHIPYARDS, DOCKS, AND PRODUCTION METHODS

(*See also Abstracts No.* 26,547, *and* 26,557)

26,549 **Development and Testing of Jig Equipment, having Contour Elements which can be Changed, for Fabricating Thin-Walled " Flat " Hull-Portions** (in German). KLOTZSCHE, K., and FALK, H. *Schiffbautechnik*, 17 (1967), p. 258 (May) [3 pp., 5 ref., 1 graph, 2 diag., 2 phot.]

Efficient jig equipment having changeable continuous-contour elements has long been available for supporting curved-plate " flat " (as opposed to " volume ") hull portions during their prefabrication, but is not suitable when the shell plate is thin (i.e. for thicknesses of about 4 to 7 mm, i.e. 0·16 to 0·28 in). In 1965–66, therefore, an East German yard, the VEB Peene-Werft at Wolgast, developed similar equipment intended for use with thin plate. The Authors describe the problem and its successful solution with this equipment, which is shown to have an economic advantage over fixed-element jigs even for series of up to 33 ships.

26,550 **Present Situation and Development Trends in Explosive Forming, and Possibilities for its Application in the Forming of Ship Plates** (in German). WIEBECK, E., and HÄNERT, M. *Schiffbautechnik*, 17 (1967), p. 253 (May) [5 pp., 18 ref., 2 graphs, 12 diag., 2 phot.]

The scope of the article indicated in the title is extended by the inclusion of brief information on electrohydraulic and electromagnetic forming (which appear, at least at present, to be suitable only for relatively small workpieces). Explosive forming is applicable to large-area workpieces, which may be of high-strength materials difficult to form by other means. Various non-shipbuilding applications are described, and it is noted that, in Czechoslovakia, double-curvature plates for Danube ships have been formed by this means. However, the general use of explosive forming for double-curvature plates calls for systematic experiments to provide basic data (theoretical analysis of the relation between the shock wave and its effect appears impracticable). Work on suitable dies is also required.

See also Abstract No. 25,911 (Nov. 1967).

MATERIALS: STRENGTH, TESTING, AND USE

26,551 **The Application of Higher-Tensile Steels in Shipbuilding: Materials, Welding, Design, and Economics.** KULLBERG, G. *Shipbuild. Shipp. Rec.*, 110 (1967), p. 483 (5 Oct.) [4 pp., 13 ref., 2 tab., 4 graphs]

This is a slightly condensed version of a paper presented to the Soc. N.A.M.E. in 1967.

A graph from the article covered by Abstract No. 26,398 (May 1968) shows that between 1956 and 1966 the average deadweight tonnage of tankers on order increased from about 33,000 to 80,000 tons, and that of the largest tanker on order from 50,000 to 280,000 tons. Concurrently, knowledge of the loads, stresses, and deformations in ships has increased; welding, manufacturing, and construction techniques have improved; corrosion protection has been improved with consequent reduction of plate thicknesses, and higher-tensile (HT) steels have been marketed. This progress has enabled ship steel weights to be reduced; e.g. the steel weights of a 70,000-ton d.w. tanker are quoted as about 15,000 tons in 1962 and 12,000 tons in 1967, 1,300 tons of the saving being due to the use of HT steel.

So far, however, HT steels have not been exploited to their full extent. In shipbuilding, strength-theory calculations are comparatively new and empirical bases are still in use. The rigidity of the ship as a whole, and those of certain local elements, are controlled by the L/D ratio; a given steel can only be used to best advantage at a certain L/D value, which will be smaller for an HT than for a mild steel. Re-design is often needed to permit utilisation of HT steels.

Under Lloyd's Register, American Bureau of Shipping, and Det Norske Veritas Rules, " material factors " (designated k, q, and f respectively) permit increases in stress levels when HT steels are used, but in no case must the maximum hull deflection exceed that for a mild-steel ship of L/D = 16. Thus, full utilisation of the advantage offered by the HT material factors is possible only when the L/D is below a certain value which depends on the steel used. With the k or q factors this L/D value

is 12·3 and with the f factor it is 11·35, for a columbium (niobium) treated steel with yield stress 36 kg/sq mm (51,200 lb/sq in), and minimum U.T.S. 50 kg/sq mm (71,000 lb/sq in).

The Author briefly discusses the welding characteristics of HT steels in relation to their equivalent carbon content (Ew). The formula for Ew adopted by Lloyd's Register in 1964 is quoted; it is now also used as a guide by the International Institute of Welding. The value of Ew as thus defined should, for good weldability, be $\leq 0·41$, a condition met by Oxelösunds Järnverk's columbium-treated (micro-alloy) steel OX 525, which has the tensile properties mentioned above. This meets the high-grade requirements of the various Classification Societies, which are tabulated. To date, ships totalling 6 to 7 million tons d.w. have been or are being built with this steel in decks, bottoms, and longitudinals. Some examples (bulk carriers and tankers) are noted, with data on steel weights and savings. A saving in steel weight, as compared with "all mild steel" construction, of 2,000 tons has been achieved in a 200,000-ton d.w. tanker of L/D 12·56 (total steel weight 26,000 tons). At present, the most economical way of using HT steel is to extend it from the bottom upwards and from the deck downwards until levels are reached where the design stress is the same in HT and in mild steel, and is the maximum allowable in the latter.

The development of OX525 has greatly reduced the differences between welding HT and mild steels; it allows a free choice of electrodes, provided these meet the strength and impact requirements. Shipyards accustomed to automatic welding and welding with high-yield low-hydrogen electrodes do not generally find any increase in welding costs, although inspection costs are higher for HT welds. The practical advantages and disadvantages, as reported by shipyards, of OX525 in relation to hull fabrication are set out; *inter alia*, the reduced plate thicknesses give a significant decrease in welding time with manual as well as automatic welding. A table gives working times (min per metre) over a range of plate thicknesses, for welding fillets and V and X butt joints with normal low-hydrogen and high-yield low-hydrogen electrodes, and also for gas cutting. The Author considers that the advantages predominate.

Future developments in the use of HT steel are discussed, with reference to the possible savings in the L.R. and D.N.V. projects for 500,000-ton tankers (see Abstracts No. 25,675 and 25,676, Sept. 1967). Increased knowledge of loading, deflections, and stress distribution in hulls may lead to the removal of the present restriction imposed by the "L/D = 16" rule. Steneroth has shown that slamming stresses do not increase with L/D (see Abstract No. 26,528, this issue). Calculations made by the Sveriges Varvsindustriförening (the Association of Swedish Shipbuilders) show that, when L/D is not a limitation and general strength criteria are applied, steels with a yield stress of 64,000 lb/sq in may be used for the entire cargo section, giving a deadweight gain (over "all mild steel" construction) of 3,240 tons in a tanker of 114,000 tons d.w.; if the yield stress is 51,200 lb/sq in, the saving is 2,070 tons. No changes in construction principles are involved. For steels of yield strength exceeding 64,000 lb/sq in, new design principles will have to be considered because of difficulties in preserving local stability. For HT steels, especially those in the 40–45 kg/sq mm (57,000–64,000 lb/sq in) yield range now contemplated, more attention will have to be paid to fatigue strength and stress

concentrations, and to associated welding problems. Recent Swedish work has shown that HT steels welded with special electrodes display higher weld fatigue strength than mild steel; this is illustrated by S-N curves for butt and fillet weldments, especially in OX525.

The Author concludes, *inter alia*, that when OX525 is used to the maximum extent possible under present Rules in the decks, bottoms, and longitudinals of bulk carriers and tankers, a steel-weight reduction of 8–10% is obtained with no increase in fabrication cost, as compared with mild-steel construction. The steel cost for such a hull is lower than for a mild-steel one if L/D is 12·3 or less. If the L/D restriction is removed, it should be possible to increase the allowable stress level to a limit set by the properties of the steel itself rather than by the methods of construction or the fatigue strength of the welds. Subject to appropriate choice of welding methods and electrodes, the use of steels in the yield range 40–45 kg/sq mm seems feasible.

BOILERS AND STEAM DISTRIBUTION

26,552 **[Prediction of and Nomogram for] Pressure Drop in [the Tube Nests of] La Mont Exhaust-Gas Boilers** (in German). Böse, D. *Schiffbautechnik*, 17 (1967), p. 308 (June) [3 pp., 16 ref., 1 tab., 3 graphs]

STEAM TURBINES AND GAS TURBINES

26,553 **Steam Turbine Propulsion—Associated Electrical Industries Successfully Introduce a Standard Commercial Propulsion Range.** *Shipbuild. Shipp. Rec.*, **110** (1967), p. 327 (7 Sept.) [4 pp., 1 ref., 1 tab., 8 diag.]

It is first argued that the advent of fast ocean-going container ships with quick turn-round has opened up new prospects for the steam turbine. Fuel consumption is only one of many factors affecting overall economy; one of the most important factors is reliability of the machinery, and another is the need for compact high-power plant. Recent advances in control and instrumentation, and in boiler design, have made steam machinery attractive from the standpoint of reliability. The Cunard Co. has ordered A.E.I. single-cylinder turbines for the two roll-on/roll-off container ships which it is to operate on behalf of the Atlantic Container Line (see also Abstract No. 26,402, May 1968). Each of these twin-screw ships is to have two type H46M sets drawn from a standard range of merchant-ship turbine sets recently developed by A.E.I.; each set will be rated at 17,250 s.h.p. (normal) and 19,000 s.h.p. (maximum).

The extensive experience (mostly naval) of A.E.I. in marine steam-turbine design since the Second World War is reviewed; the two single-cylinder sets in the standard range are largely based on their naval counterparts. The advantages of single-cylinder over cross-compound machinery are simplicity of construction, operation, and maintenance, lower first cost, and greater compactness (especially in the transverse direction). At moderate powers these considerations may outweigh the lower (by about 3½%) thermal efficiency. Some basic design principles are mentioned.

The standard range comprises three cross-compound units (2H61M, 40,000 s.h.p.; 2H51M, 25,000 s.h.p.; 2H46M, 18,000 s.h.p.) made up by selection from two H.P. and three L.P. turbine sizes; and two single-cylinder designs (H46M, 18,000 s.h.p.; CH28M, 10,000 s.h.p., no astern turbine). Normal inlet steam conditions are 900 lb/sq in and 950° F, except for CH28M, where they are 600 lb/sq in and 850° F. Increases in inlet conditions are possible with only minor modifications; in this case the maximum outputs already noted are correspondingly increased.

The H46M design is described in some detail. The alloy-steel (Ni-Cr-Mo-V) rotor is of the integral type; the ahead section has 14 single-row impulse stages and the astern section forward of it a two-row Curtis wheel followed by a single impulse stage. Expansion is in the aft-to-forward direction through the ahead turbine, and sternwards through the astern one; there are common openings for ahead and astern exhaust. The initial stages of the ahead section have Mo-V steel blading and diaphragms; the remaining blades are of stainless steel, with Stellite erosion shields in the last row. Except for the last two ahead rows, all rotor blading is shrouded. The inlet nozzles are fully machined from Mo-V steel, and are fitted in thermally-flexible nozzle boxes of the same material; a group of overload nozzles is provided.

The ahead and astern casings are single, the H.P. part of the ahead one being an Mo-V steel casting; they are rigidly supported from the vertical joints at their exhaust ends. These vertical joints are themselves rigidly borne by two strong longitudinal beams at half-height. The ends of the beams are connected to the exhaust casing and outer support structure through flexible plates which ensure that movements of the external parts mentioned cannot impose bending moments on the beams. The beams are, moreover, entirely surrounded by exhaust steam to avoid thermal deflections. The astern casing is cantilevered from its vertical joint, and the foremost bearing is carried on an extension of its lower half in a manner allowing for differential expansion between casing and bearing housing. The after end of the ahead casing is supported from a separate pedestal through palms at half-height and a longitudinally-sliding vertical key at its bottom; this pedestal is itself free to slide on supports fixed to the two beams. This arrangement, in which the casings and bearings are carried entirely by flexibly-mounted beams, ensures proper alignment and clearances under all operating conditions. The condenser is of underslung type, with fore-and-aft tubes.

The manœuvring valves are hydraulically actuated, the pump being driven off the end of the first reduction. A cam-operated feedback device ensures an approximately linear relation between valve-controller movement and propeller speed. The reduction gearing is of double-helical dual-tandem articulated type, with nitrided pinions. The amalgamation of A.E.I.'s gearing establishments at Manchester and Rugby is noted, as are the firm's successful experiments with nitrided pinions and through-hardened wheels at K factors up to 600 (over 1,000 has been achieved with both components surface-hardened).

The support system of the single-cylinder machinery is also adopted in the L.P./astern cylinder of the cross-compound sets; many other design features are common to both types, but the cross-compound machinery drives through single-tandem articulated gearing. The article includes sectional drawings of the H46M and the 2H61M turbine designs.

26,554 Application of Gas-Turbine Combined Cycle to Marine Steam-Turbine Plant. MATSUOKA, H., YAMATE, S., and TSUSHIMA, K. *Mitsubishi Heavy Industries Technical Review*, 5 (1968), p. 1 (Jan.) [9 pp., 4 tab., 8 graphs, 9 diag.]

The continuing increase in the size of tankers demands a constant search for increased efficiency in their propulsion plants. The Authors refer to improvements which have already been incorporated in the Mitsubishi MTP, MS, and MR plants which were described in Abstract No. 25,819 (Oct. 1967).

Further work has now been completed on the design of steam and gas turbine combined-cycle plants to be designated MGS (non-reheat) and MGR (reheat). The plants comprise a conventional 60 or 100 kg/sq cm (853 or 1,422 lb/sq in) steam turbine as the main turbine, and an oil-fired 710° C (1,310° F) gas turbine which drives the main generator and main feed pump independently of the propeller shaft. Exhaust gas from the gas turbine is used as a combustion medium for the main boiler. A flue-gas/air heater is incorporated. Excess electrical output from the generator is fed to a motor providing supplementary shaft power.

The article gives a full account of these schemes, including estimated performance data. The gas-turbine combustor burns washed heavy oil. MGR plants are expected to have a fuel consumption of about 170 g (0·35 lb)/s.h.p.-hr, and to be flexible in operation.

26,555 The Expansion Turbine. ANDERSON, J. H. *A.S.M.E., Paper No. 65-WA/GTP-13, presented* 7–11 Nov. 1965 [4 pp., 8 ref., 3 graphs, 1 diag., 1 phot.]

Properly, an expansion turbine is any turbine in which the working fluid increases its specific volume (thus excluding hydraulic turbines). The present paper concentrates on turbines working with unusually small temperature differences, or with unusual fluids. A chart is first given showing the ranges of " head " (defined as ft-lb of energy produced per lb of throughput) and specific flow (lb per h.p.-hr) covered by different categories of hydraulic and expansion turbines. Water meters are at the lower, and steam turbines at the upper, end of the " head " scale; it is noted that only steam and some combustion-gas turbines operate in a region requiring more than one stage for good efficiency, and that multi-stage high-expansion turbines have inherent drawbacks. The factors influencing the diameter of single-stage turbines are then considered; one of the most significant is fluid density. This is illustrated by some data showing why a single-stage propane turbine is practicable, and a water-vapour one impracticable (even with two stages), for " sea thermal power " (a scheme for obtaining electric power from the temperature difference between the surface and lower strata of the sea).

Various applications of expansion turbines are noted, and some are briefly discussed. These include: " gas-expansion " refrigeration cycles; the use of a cycle employing a hydrocarbon, halocarbon, or other heavy fluid at the condensing end of a large steam-power plant; closed-cycle halocarbon turbines for vehicles; and a " turbine comfort " halocarbon cycle capable of simultaneously heating one part of a large building and cooling another.

The variety of fluids used in expansion turbines is rapidly increasing. It is therefore desirable that the A.S.M.E. should draw up a Test Code

giving sound methods for relating performance data obtained during manufacturer's tests with a particular fluid to the performance to be expected in service with a different fluid. Such a Code is already available for compressors.

DIESEL AND OTHER I.C. ENGINES

(*See also Abstract No.* 26,559)

26,556 **Design Methods and Development of Medium- and High-Speed Oil Engines.** HOWE, A. G., and WATSON, H. *Inst. Mar. E., paper presented* 11 Apr. 1967 [16 pp., 3 ref., 1 tab., 6 diag., 7 graphs, 10 phot.]

This paper deals separately with two kinds of Diesel engine produced by the Ruston Paxman Group, namely, the high-speed engines of Davey Paxman & Co. Ltd, and the medium-speed engines of Ruston and Hornsby Ltd.

High-Speed Engines. Davey Paxman produce engines ranging in power from 4 h.p. to 8,000 h.p. The type chosen by the first Author to illustrate the design and development methods used by this firm is the Paxman Ventura. This is a V-type having a bore of 7¾ in and a stroke of 8½ in; it is available with 6. 8, 12, or 16 cylinders, giving engine ratings of 500–2,400 b.h.p. at 1,500 r.p.m. A 20% overload can be taken for short periods in such applications as high-speed marine craft. The b.m.e.p. increases from 160 lb/sq in at the continuous rating to 200 lb/sq in at the marine overload. The specific weight at full-load rating is 6·3 lb/b.h.p.

The Ventura was developed from successful earlier Paxman V-types, the aim being to increase the output by some 50% without increasing the weight or overall dimensions. This aim was achieved mainly as a result of the following design features:—

- A welded-steel engine housing comprising the crankcase and cylinder blocks.
- The enlargement of both bore and stroke to give a 23% increase in cylinder capacity within the same overall length.
- The use of high-grade materials, together with attention to surface finish and hardness.
- The use of high-pressure turbocharging and intercooling, together with design for higher cylinder pressures.

The need for accessibility was also a guiding principle in the design. The cylinder heads can be removed individually without removing the air or exhaust manifolds; parts needing attention, such as injectors, fuel pumps, etc., are on the outside of the engine, whereas the exhaust system is within the V; the big-end bearings are of the thin-wall lead-bronze type and can be dismantled for examination through the crankcase doors; the main bearings and crankshaft can be inspected by using specially designed tools and jacks.

The problem of oil leakage, which can be troublesome in high-speed engines, received careful attention. Among other measures, recessed synthetic-rubber bushes about ½ in long have been adopted to seal the various ferrule connections between cylinder heads and cylinder block.

The Author describes in some detail the design of the welded crankcase, the crankshaft, the cylinder heads and pistons, the connecting rods, and the cooling system. The surface finish of the crankshaft must be good, the

pins and journals being held to eight micro-inches. The angle of the oil hole in the crank pin was also found to be important, and it is now at right angles to the pin in accordance with the result of fatigue tests made by the National Engineering Laboratory. The piston is a single-piece aluminium casting, which has been found satisfactory for b.m.e.p. up to 200 lb/sq in.

The Ventura has been adopted as the Standard Range II by the Royal Navy, and has also been installed in various non-naval marine craft.

Medium-Speed Engines. Since the mid-1950s the power output (per unit piston area) of medium-speed Diesel engines has increased rapidly and is today more than three times what it was ten years ago. This rate of increase, which has been achieved without any sacrifice of reliability or life, is largely due to the use of improved design techniques by specialist design teams, and to the introduction of development testing as a logical step in the design process.

To illustrate how these techniques are applied, the second Author describes the development of the Ruston AT engine; this has a bore of 12½ in and a stroke of 14 in, and is made in in-line form with 6, 8, and 9 cylinders and in V-form with 12, 16, and 18 cylinders. All have turbochargers and intercoolers. Design work began in 1956 and, after extensive development on prototype units, the engine was first released for service in 1959 at a continuous rating of 160 lb/sq in b.m.e.p. at 500 r.p.m. In 1962 this rating was increased to 200 lb/sq in b.m.e.p. In 1965, the speed was increased to 600 r.p.m. and the b.m.e.p. to 205 lb/sq in in 6 and 9 cylinder combinations. It is expected that this engine type will eventually achieve a continuous b.m.e.p. of 300 lb/sq in at 750 r.p.m. The four successive ratings correspond to 180, 225, 278, and 500 h.p. per cylinder. The largest unit in the present range is an 18-cylinder engine developing 5,000 h.p. at 600 r.p.m. An account is given of the main mechanical features of the AT engine, including the materials and types of construction used for the crankshafts, connecting rods, bearings, pistons, cylinder liners, cylinder heads, and fuel-injection system.

The Author specifies the main problems in designing a new engine, and describes how they were tackled in the case of the AT range. The complexity of the task is illustrated by a diagram showing 141 inter-related steps in the initial design process. In general, from four to seven years may elapse between the decision to proceed and the production of an engine capable of satisfactory service operation. The main types of design problem are (*a*) performance prediction; (*b*) problems associated with purely mechanical loading; (*c*) problems where a component is subject to combined mechanical and thermal loading. The solution of problems of type (*a*) has been greatly assisted by computers, which have been used to assess the effects of different combinations of variables such as turbocharger pressure ratio, fuel-injection rates, maximum cylinder pressure, and others which the Author lists. The results, while not quantitatively accurate, were useful in establishing performance trends with the different variables, thereby providing a basis for parameter variation in actual engine tests. The results of these tests were fed back into the computer program, which then gave more accurate predictions. A table shows the good accuracy with which measured values of various performance parameters can now be predicted by the computer.

Dealing with mechanical problems, the Author points out that few

engine components can be designed without experimental verification, in which the use of models can be most helpful. Rubber models, which can be quickly made and altered, readily indicate stress-concentrations or excessive deflections, and the best way of utilising the material. Such models were used in designing the bedplate of the AT engine. Scale perspex and epoxy-resin models were also used with success to measure stresses and deflections. Among the specific problems discussed is the prediction of bearing loads; it is shown how bearing failures from which a certain type of low-rated engine had suffered were overcome. From a drawing of the load diagram relative to the crankpin it was evident that a modification of the balance-weight system would be effective.

As an example of problems of type (*c*), the Author discusses the design of the cylinder head of the AT engine. To achieve the high output required, a four-valve head arrangement was necessary and, since the engine was to be suitable for high-viscosity fuels, it was decided to reject valve cages. Using theory and practical experience from past engines, the designers produced prototype units for temperature-distribution and stress measurements, from the results of which further designs were made. The method of making the temperature measurements, some 7,500 in all, is briefly described (see also Abstract No. 23,000, May 1965).

LAYOUT AND INSTALLATION

26,557 **Possibilities for the Application of the Modular System to Ship Cabins** (in German). WULSTEN, U., and WULSTEN, F. *Schiffbautechnik*, 17 (1967), p. 378 (July) [5 pp., 7 ref., 4 tab., 1 diag., 4 phot.]

In the proposed system here described, cabins would be constructed (in the ship) from parts selected from standardised series produced in large quantities. The system, including cabin walls, floors and ceilings, furniture, and sanitary fittings, is based on a 200-mm module (i.e. on dimensional steps of 200 mm). All static and dynamic loads are taken by standardised box-section pillars (held by top and bottom pieces welded to the ship structure); the wall sections, etc., are attached to these by a combination of interlocking and bolting. The wall sections are of double-panel type, filled with a non-inflammable insulant. Possible cabin arrangements are discussed, and illustrated by photographs.

26,558 **Simplifying the Design Process by Magnetic-Rubber Techniques** (in German). SCHRÖDER, E. *Schiffbautechnik*, 17 (1967), p. 490 (Sept.) [4 pp., 2 graphs, 3 phot.]

A description is given of a two-dimensional method of representing such assemblies as systems of pipes, ducts, and cables while designing their layouts. The technique can also be used for machinery layouts, possibly as a preliminary to constructing a three-dimensional model.

The method employs pieces of magnetic rubber 1·8 mm (0·07 in) thick; the material is supplied in East Germany under the trade-name "Manigum", and is available in several colours. The pieces are placed on a sheet-steel panel; their magnetic properties hold them in position, and their elasticity is useful when fitting the pieces together. The pieces can themselves be in the form of scale models of components, or can be used as bases carrying standard symbols representing the various components. A scale of 1 : 25 has been found satisfactory for piping and

similar systems, but 1 : 50 should be sufficient for such work as accommodation and machinery layouts.

An outline drawing showing the " given " parts of the area concerned is stuck on to the panel as a foundation for the layout which is to be designed, and the design work can then proceed in much the same way as usual except that no drawing or text is necessary. Photographic enlargements of the finished design replace conventional drawings of the completed layout.

The two main applications of the method are scale representations, as indicated above, and functional diagrams (which need not be to scale). In the design of a 12,000-ton d.w. cargo motor-ship, the method was used for the simpler piping layouts (three-dimensional models were used for those in the machinery spaces), functional diagrams for the machinery, machinery layouts, wiring diagrams, accommodation layouts, and equipment layouts.

Compared with making conventional drawings, the method has been found to require 20 to 25% less time. The article includes further information on these techniques, as used in the VEB Warnowwerft at Warnemünde, together with a summary of their advantages and their comparatively few drawbacks.

MARINE POWER INSTALLATIONS (GENERAL)

26,559 **Diesel Propulsion on Great Lakes Ships.** SPOONER, C. W., and TRIPP, C. E. *A.S.M.E., Paper No. 66-DGEP-10, presented* 24–28 Apr. 1966 [21 pp., 12 tab., 3 graphs, 3 diag., 7 phot.]

With two exceptions, all the Great Lakes bulk carriers built up to 1960 were steam-driven. During the last part of the 1939–45 war and immediately afterwards, many of the 600-ft class ships had become old enough to be troubled with increasing boiler maintenance, and many of them were re-equipped with more modern boilers. Since the 1950s increasing numbers of these old vessels have been re-engined, some with steam-turbine machinery and some with Diesel engines.

Since 1960 no new U.S. Great Lakes bulk carriers have been built, but, owing to more favourable economic conditions, 24 new Great Lakes and Seaway bulk carriers have been constructed in Canada, as well as six self-unloaders and nine package freighters. Most of the bulk carriers have been of the maximum size (730-ft length, 75-ft beam) allowed by the locks. Many of these new vessels, particularly those built during the last two years, are Diesel-propelled, some by medium-speed, geared multi-engine systems (see also Abstract No. 23,330, Aug. 1965) and others by single slow-speed direct drive.

The Author discusses the relative economic advantages of steam-turbine and Diesel propulsion, not only for new vessels of the 730-ft class but also as they affect the repowering of the older 600-ft class. Many of these latter vessels, despite being 40-60 years old, have hulls in relatively good condition but still have the old hand-fired coal-burning Scotch boilers.

The principal particulars of the new 730-ft class are:—

Length, o.a.	730 ft
b.p.	712 ft
Breadth, moulded	75 ft

Depth, moulded, to spar deck . .	39 ft	
Draught, moulded, summer . . .	26·5 ft	
Displacement at this draught (fresh water)	33,920 tons	
Gross tonnage	15,800	
	Turbine	*Single Diesel*
S.h.p., normal	9,000	9,000
R.p.m.	108	115
Speed, loaded (knots)	16·8	16·7
Estimated cost in 1966 (U.S. dollars)		
Ore and grain carrier, built in Canada	7,220,000	7,000,000
built in U.S. .	9,000,000	—

In his comparative study, the Author assumes that the turbine-driven ship is oil-fired. Of the 28 ships of the 730-ft class so far built only one burns coal. At Canadian prices for coal and oil fuel, oil firing shows an economic advantage when maintenance and man-power for operation are also taken into account. The Author believes that only fossil fuels need be considered as likely to be used in the foreseeable future, and that gas-turbine propulsion has yet to become commercially attractive.

A further assumption made for the new 730-ft ships is that they have three 500-kW Diesel-driven generators for their electrical power. Both the U.S. and Canada seem to favour steam-turbine driven generators, but the Author points out that, in the 500-kW range, a Diesel set costs about half as much as a steam set, and has operational advantages as well.

Cost figures are evaluated for a voyage of 1,616 miles (the reason for selecting this mileage is not explicitly stated). The operating costs for one year are given as $155,000 for the steam-turbine and $119,000 for the Diesel drive, the first costs of these two types being $1,490,000 and $1,272,000 respectively. The Author points out that the assumed 9,000 s.h.p. of these vessels is rather larger than is justified economically, but it has now been adopted for a large number of the ships; 7,500 s.h.p. would be more appropriate and would cost some $250,000 less.

In his analysis of the effects of repowering the 600-ft ships (whose principal particulars are tabulated), the Author considers a simple change from the old boilers to modern oil-fired boilers, and a complete repowering with new Diesel plant. For a particular iron-ore trip from Superior, Wisconsin, to Ashtabula, Ohio, he estimates that a gain of four round trips per season could be expected with the repowered vessel when the existing 2,050-i.h.p. triple-expansion steam plant is replaced by a medium-speed geared-drive Diesel of 3,200 b.h.p. The reduction in operating cost per season is $42,000 for the reboilered ship and $76,150 for the repowered one; as the repowered ship can carry more cargo than before (the Diesel plant being some 420 tons lighter than the original) the gain in net revenue per season for this case is $200,750. The estimated cost of reboilering is $800,000, and of repowering $1,250,000, i.e. 19 and 6·23 times the net revenue gains.

HEAT TRANSFER AND INSULATION

26,560 **An Optimum-Design Method for the [Thicknesses of] Insulation in Ships' Refrigerated Spaces (in German).** CLASEN, E., and SCHOLZ, K. *Schiffbautechnik*, 17 (1967), p. 437 (Aug.) [3 pp., 4 ref., 1 tab., 2 diag.]

Because of the difficulty of establishing the appropriate thicknesses of

insulating material required on the various surfaces for the optimum insulation of refrigerated spaces, expenditure on design forms a large part of the cost of the insulation system. The Authors therefore propose a time-saving direct method for calculating the various thicknesses of insulation appropriate to the different bulkheads and other partitions. Heat bridges can be taken into account.

When the type of insulating material and the required temperature differences for the various surfaces have been decided, the method can be used for any refrigerated space provided either the average heat-flux density or the heat-transmission coefficient (K) is known. If the insulation material and temperature differences have been chosen on an economic criterion, the results of the calculations will be the cheapest solution.

The method is explained from first principles. The University of Rostock has compiled computer programs for the solution of the equations involved in the calculations.

26,561 **A Method for Determining [by Simple Direct Calculation] the Economical Thickness of Thermal Insulation for Ship Compartments (in German).** HELLER, K.-H. *Schiffbautechnik*, 17 (1967), p. 264 (May) [4 pp., 4 ref., 1 tab., 2 graphs]

The method presented takes account of heat bridges; heat leakage into or out of the space; the cost of heating or refrigeration; capital repayment and maintenance and repair costs; and the costs associated with lost volume in holds, etc.

26,562 **Investigation of the Heat Exchange in the Cylinder of a Low-Speed Marine Engine, Operating with Various Cooling-Water Temperatures** (in Russian). VINOGRADOV, T. L. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 75 (1966), p. 15 [10 pp., 5 ref., 3 tab., 1 graph, 4 diag.]

This article describes and discusses temperature measurements on the cylinder head, liner, and piston of a B. & W. 574-VTBF-160 engine, working with cooling-water temperatures (at outlet) of 60°, 70°, and 85° C (140°, 158°, and 185° F).

AUXILIARY EQUIPMENT AND MACHINERY

26,563 **Investigation, by Mathematical Simulation, of the Operation of [Centrifugal Condensate and Feed] Pumps with Unstable Pressure Characteristics** (in Russian). VORCHAKOV, M. T., ZOMMER, G. V., and others. *Sudostroenie*, No. 7 (1967), p. 26 (July) [4 pp., 4 ref., 6 graphs, 2 diag.]

Most of the centrifugal condensate and feed pumps currently used in marine propulsion plants are inherently unstable at low throughputs; this can lead to undesirable and possibly dangerous fluctuations of pressure and flow. This article describes, and presents the results of, an electronic-analogue investigation of the phenomenon. The accuracy of the technique was established by comparison with laboratory measurements on a simple pumping circuit. Means of reducing the fluctuation amplitudes are suggested.

26,564 **The Design of Electrical Systems in Ships.** COOPER, C. B. *Inst. Mar. E., Marine Electrical Engineering Section, paper presented* 1 Nov. 1967 [7 pp., 4 ref., 3 diag., 9 graphs]

The Author specifies the data required for the design of a ship's electrical system, and discusses methods of generation, voltages, fault currents, earthing, switchgear and fuse gear, and dynamic performance of the system. He deals almost exclusively with A.C. generation, pointing out that D.C. is now used only on the smallest ships and never when the total power generated exceeds 1,500 kW.

The initial planning of a ship's electrical system has to be based on past experience to indicate the probable size and types of loads and service requirements. The desirable design data include a knowledge of the length of time for which each item of plant is likely to be used and the reliability required for its supply. The latter information is important since special steps must be taken to safeguard the essential items, and this can involve the temporary disconnection of less important items. The normal maximum demand and the maximum essential demand are factors that influence the choice of main and auxiliary plant and their sizes. Large generators are more economical in operation, cost less, and require less space in relation to their output than smaller sets, but the consequences of outage are more serious.

Generation. A frequency of 60 or 50 c/s is commonly used since these are the normal frequencies of shore supply. The higher frequency requires smaller and cheaper machines than the lower because, for given dimensions, power output is proportional to frequency. The Author suggests that a frequency of 80 to 100 c/s might give useful savings of weight and space.

Vessels driven by steam turbines generally have turbo-generators, but stand-by Diesel sets are carried as well for raising steam initially as shore supplies are not always available. When available, shore supplies are usually taken at about 440 V. It is possible to feed a 60 c/s system at 50 c/s if the voltage is reduced in the ratio 5/6, but a 50 c/s system should never be fed at 60 c/s.

With a D.C. system a generator can be driven from the propeller shaft, the excitation being varied to maintain constant output voltage irrespective of shaft speed. The development of compact thyristor rectifier/inverters may enable A.C. generators to be similarly driven.

Fault currents. The choice of switchgear and cables depends not only on the load current to be carried, but also on the current passing in the event of a solid three-phase short-circuit. The characteristics of fault currents are explained, and a method of calculating them is given in an appendix.

The mechanical strength of a circuit breaker is important when it is closed on to a fault. The short-circuit rating is very important for 440-V systems: for a 250-h.p. motor the cable size at 440-V based on the continuous load rating would be 37/·083, but if the fault level were 50 MVA the size required would be 61/·103.

System voltage. The 415–440 V range is the most common on ships, but 3·3 kV is also being used. The choice depends on fault level and cost. The higher voltage reduces the destructive effects of large fault currents, but the effects of shock are much more severe and much greater care is

needed to protect personnel. As regards cost, the economic choice depends largely on the cable lengths required.

System earthing. Present marine practice is to operate 440-V systems with an insulated neutral and 3·3-kV systems with the neutral earthed to the ship's structure through a resistor. The first scheme has the advantage that when an earth fault develops the plant need not be disconnected from service, but dangerous over-voltages can arise, especially with "intermittent" faults. Also, it is difficult to locate earth faults and clear them quickly. Dangerous over-voltages are much less likely to occur with a resistor earthed system, but earth-fault currents are larger and may cause damage or be dangerous. The Author considers that there are arguments for using high-resistance earthing in both 440-V and 3·3-kV systems.

Choice of switchgear and fusegear. Fusegear has advantages over switchgear in respect of speed of fault clearance, size, and cost. Switchgear, however, provides good discrimination and enables faulty plant only to be disconnected. Both devices are therefore used as a rule. Various possible combinations are described.

Dynamic performance of the system. The response of the system to variations in demand is important. The Author discusses the effects of the following types of disturbance:—

(1) Single disturbances which are predictable, such as those that occur when starting motors.

(2) Single disturbances which are not predictable, such as those caused by system faults.

(3) Continuous fluctuations such as those due to a cyclical load or to hunting of generators resulting from maloperation of a governor or voltage regulator.

In short concluding sections, the Author discusses tolerable voltage variations, and advocates the use of digital-computer methods to replace the rule of thumb methods hitherto used, albeit with considerable success, in the early design stages of marine electrical systems.

26,565 Electric Motor Protection. DRING, E. *Inst. Mar. E., Marine Electrical Engineering Section, paper read* 1 Nov. 1967 [12 pp., 1 tab., 17 graphs, 1 diag., 2 phot.]

26,566 Transient Performance of A.C. Generators. OGLE, H. R. *Inst. Mar. E., Marine Electrical Engineering Section, paper read* 1 Nov. 1967 [8 pp., 1 tab., 6 diag., 12 graphs]

26,567 Distribution of the Current in a Hull [when Serving as a Return or Neutral Conductor], with D.C. and with A.C. Electrical Installations (in German). PULOW, O. *Schiffbautechnik*, 18 (1968), p. 142 (Mar.) [4 pp., 3 ref., 1 tab., 3 graphs, 3 diag.]

AUTOMATION, INSTRUMENTS, AND CONTROL DEVICES

(*See also Abstracts No.* 26,519 *and* 26,542)

26,568 A Cost-Effective Approach to Marine Automation. SAMUELS, B. M., and EAST, A. M. *Special Survey of Marine Automation and Remote Control, published by Motor Ship*, Dec. 1967. p. 39 [2 pp., 2 tab., 2 diag.]

The present method of selecting marine plant and equipment is based

on concepts that originated during the early days of steam; the prime consideration is to minimise operational costs as represented by fuel consumption and depreciation of capital. To-day, it is essential to aim also at efficient use of labour, and the whole ship should be regarded as an integrated system. A logical development is, therefore, to apply systems engineering techniques which will take account of the inter-relation of various methods of reducing running costs so as to optimise the system as a whole.

The Authors take as an example a general-cargo ship with the following main particulars:—

Length, b.p.	500 ft
Breadth	65 ft
Draught	30 ft
Corresponding deadweight	14,500 tons
Loaded service speed	14 knots
Propulsive power (Diesel, 3,500 sec. fuel)	6,000 b.h.p.

The average complement for such a ship on present manning scales is 30–31 officers and men. The tasks of those connected directly with the operation of the ship reach peaks during short periods such as when leaving or entering harbour or changing speed or direction. When the traditional system of operation was evolved, the available techniques could not reduce the amplitude of the peaks, and the manning scale had to be based on the efficient handling of the ship during these periods of peak loading.

With the correct application of modern process instrumentation, the peak effort can be drastically reduced if the main consideration is economy of manpower instead of economy of materials. If these concepts are applied to the ship under consideration, it is estimated that the complement can be reduced by 12, which will save about £500,000 in 20 years. The cost of the new instrumentation which must be installed to achieve this saving is about £50,000. This instrumentation will allow for centralised bridge control of manœuvring with fully-automatic follow-up of main machinery, 14/16 hours' unattended running of machinery in normal underway conditions, and complementary centralised machinery control from a separate location.

Further economies can be obtained by the more flexible use of labour in non-specialist tasks. The use of Venn diagrams to achieve this is explained. It is suggested that, ultimately, the complement for the ship considered could be reduced to 10 or 11 men, for whom a suggested duty roster is tabulated. Apart from the master, who has to be available continuously throughout the 24 hours, and the " hotelier ", who is shown as being on part-time duty between 4 a.m. and 8 p.m., no crew member works for more than two four-hour spells per 24 hours.

26,569 Design Study for Unmanned Engine Room for Stern Freezer Trawlers. HATFIELD, M., and SWITZMAN, J. *Int. Shipbuild. Progress*, 15 (1968), p. 140 (Apr.) [9 pp., 1 diag., 1 phot.]

Despite the recent technological advances in equipment and improved standards of crew comfort that have occurred in distant-water trawlers during the last 20 years or so, there has been a steady decline in the

recruitment of both deck crew and engineers; and it is becoming increasingly difficult to find men willing to go to sea on any but the most modern and well-found fishing vessels.

For this reason, the Industrial Development Unit of the White Fish Authority has been investigating the possibility of drastically reducing the responsibilities, and hence the numbers, of the ship's engineers. A design study was put out to tender for a manpower-saving automation system, and the English Electric Company, Power and Marine Division, was selected to carry it out. The vessel chosen for the study was the *Criscilla*, one of the fleet of distant-water stern freezer trawlers then being built for J. Marr & Sons by Hall, Russell & Co. Her leading particulars are:—

Length, o.a.	185·5 ft
Breadth, moulded	36 ft
Gross tonnage	1,000
Propulsive power	1,680 b.h.p. at 400 r.p.m.

The engine is a seven-cylinder Mirrlees KSSMR, coupled to an R-type MWD gearbox, and driving a Liaaen controllable-pitch propeller. A Spanner exhaust-gas/oil-fired boiler generates 1,000 lb of steam per hr. and there is a 3 × 60 h.p. two-stage refrigeration plant by L. Sterne & Co. Power for the auxiliary equipment, which includes motor-driven sea-water, oil, fuel, and fresh-water pumps, purifiers, etc., and deck machinery typical of the class, is provided by a 220-kW auxiliary generator and a 240-kW trawl-winch generator (both driven off the gearbox), together with standby and harbour Diesel sets.

Such a vessel would normally carry a chief engineer, two second engineers, and two greasers. From an examination of their duties it was decided that the only way to achieve a worthwhile staff reduction without adding materially to the work of any member of the crew would be to eliminate completely the permanent engineers' watch, which occupies the full time of two men.

After a period spent in collecting voyage data and in discussions with owners, builders, and equipment suppliers, mainly in order to meet the W.F.A. specification that the main items of the plant would have to be without service or attendance for the 6–8 weeks of each voyage, final proposals were put forward. The basic conclusion is that it is feasible, by the use of automation, to have an unmanned engine room in this type of vessel and to retain a reasonable work load for the engineering staff remaining aboard (two engineers and one greaser); no other crewmen need carry any significant extra duties.

The proposals are basically that (*a*) all watchkeeping and control functions of engine-room machinery should be centralised in one control room, and (*b*) a supervisory system be substituted for the functions normally performed by the engineer watchkeepers. The supervisory system should provide automatic watchkeeping for all machinery and plant items aboard, including alarm annunciation in appropriate parts of the ship; it should also provide automatic or remote operation of certain items of plant where delay could have serious effects.

With such a system, routine duties remaining for the engineers on board would include preparation of the main engine for cold start; salt-water ballasting; limited operations in connection with fuel transfer, oiling, and greasing; cleaning fuel and oil purifiers if manually cleaned

types are fitted; cleaning and occasional visual inspection of machinery; and emergency repairs. The greaser or one more engineer could possibly be dispensed with if certain extra operations were included in the automatic system, such as main engine start from cold, main engine lubrication, and cleaning of purifiers.

The functions of the automatic supervisory system are summarised, and a description is given under the following headings: output typewriter, display, and control facilities; plant input state scanner; plant input analogue scanner; timing and interrupter unit; plant control sequences; the consoles and their siting; extensions of the system into other parts of the ship.

The W.F.A. and the English Electric Company are now considering the application of the proposed system to one of a new class of stern trawlers being built by Ferguson Bros for Thomas Hamling Ltd.

26,570 **Automation in Tugs—Applications and Results.** CAMPBELL, A. J. S. *Special Survey of Marine Automation and Remote Control, published by Motor Ship*, Dec. 1967. p. 35 [2 pp., 1 diag., 2 phot.]

In 1962 the single-screw tug *Arrow* of the Manchester Ship Canal Co. was converted to Diesel drive and equipped with full engine control direct from the telegraphs in the wheelhouse. As a steam tug she carried three engineering crew; this number has now been reduced to one, and the Company's objective for future tugs is to operate without engineering crews at all, merely having one man in attendance in the engine room. Any skilled engineering assistance needed will be obtained from one of several workshops along the Canal.

The control system adopted for the *Arrow* is such that a push button pressed in the wheelhouse starts the main engine after starting the necessary auxiliaries such as main alternator, compressor, oil priming pump, steering pump, etc., in the correct order. It is divided into two sections—one for the main engine and one for the main alternator; the latter can be started by itself when desired. If the engine fails to start at the first attempt the sequence begins again at the stage of failure until three attempts have been made. Failure to start after the third attempt initiates an alarm and shuts the system down. Stop buttons reverse the sequence.

A description and sequence diagram are given of the system, which is made up of standard relays of the plug-in type except for the timers, which have pneumatic relays. No mechanical linkages are used between the telegraph and the engine, the control being entirely electrical. Although this scheme has given satisfactory service, the Author now prefers the more recent power-assisted mechanical linkage system since it is not put completely out of action by an electrical failure. This system has been fitted to two tugs built since the *Arrow* conversion.

Five years' experience with the converted *Arrow* has shown very few faults or failures, and of those that have occurred most have been due to human error. In the early stages, one particular fault was traced to the effect of vibration, and was rectified by the substitution of a different type of relay.

Vibration, high ambient temperatures, and oily engine-room atmospheres are the main problems to be considered in marine-engine automatic-

control systems. Static equipment using logic elements would seem to be preferable to relay systems for a number of reasons.

26,571 **Improved [Semiconductor Control] Circuits for Automated Electric [Auxiliary] Drives** (in Russian). AVIK, YU. N., and SERZHANTOV, V. V. *Sudostroenie*, No. 5 (1967), p. 29 (May) [4 pp., 3 tab., 1 graph, 8 diag.]

The use of contactors in automated systems involving the control of electric motors makes the equipment cumbersome; also, the large number of contacts reduces reliability and necessitates careful maintenance. There is consequently a tendency to adopt contactless devices; this is quite feasible on the basis of modern developments in semiconductor techniques, and results in more compact equipment giving better protection and control. The Authors describe several such semiconductor systems in some detail; circuit diagrams are given in each case.

The first example is an automatic control system for a Freon refrigerating installation, suitable for provision stores. Features of this scheme are free selection of the " basic" cooled chamber with the governing thermal relay, and the use of a semiconductor timing relay, instead of the usual type, for ensuring that compressor and pump motors will start only when there is enough pressure in the bearing-lubrication system. Circuit diagrams are given of the timing relays for D.C. and for A.C. supply. The delay is shown graphically as a function of a variable resistance in the circuit. A table shows the advantageous features of the semiconductor relay (delay-time, range, dimensions, weight, cost) in comparison with conventional timing relays.

SOLAS requirements state that steering-gear electric motors should have short-circuit but not overload protection. It is therefore necessary to provide visual and acoustic overload alarms. A semiconductor scheme, as used for the three-phase motors of electro-hydraulic steering gears in some harbour icebreakers and in vessels of the *Amguema* class (436 ft o.a.), is described. Each indicating lamp shines steadily to show that the motor concerned is on line, and is switched over to a flashing circuit when a thermal relay gives an overload signal. A variable resistance allows the brightness of the lamps to be adapted to the illumination conditions in the wheelhouse.

Ships now have numerous automated pump and compressor drives controlled by impulses from pressure-relays, level relays, etc. As a rule they operate for long periods unattended. It is generally considered that, for motor overload protection in such cases, a thermal relay on the starters is adequate, and that in practice overloading cannot occur because the motors usually have a certain power reserve. A conventional control circuit for such a drive, with the addition of overload warning devices, is shown. Such a scheme will only work reliably if the thermal relays are not self-resetting. Three modified circuits which avoid this drawback are then described. The variant which should be used in a given case depends on the application of the motor, on whether or not a watch is kept nearby, and on other operating conditions.

The examples described show that it is necessary, in designing such circuits, to consider very carefully the actual operating conditions and requirements of the given drive. Semiconductor techniques should be applied more extensively.

26,572 **Rotating Vibration Test of Turbine Blade by Telemetering System.** ARIKAWA, S., and MATSUOKA, H. *Mitsubishi Heavy Industries Technical Review*, 5 (1968), p. 10 (Jan.) [7 pp., 10 ref., 1 tab., 5 graphs, 16 diag., 6 phot.]

The system here described comprises a battery-operated FM transmitter which is mounted on the turbine rotor, and a receiver fitted on the stator. Strain gauges attached to the rotor blades are connected to the transmitter, and the resulting signals pass to the receiver via antennae and are tape-recorded.

The development and testing of the system are described in detail. In the course of the experiments, the natural frequency of a 28-in blade in the final stage of a 350 MW turbine was measured during rotation, and good results were obtained. The frequency stability of the transmitter under varying conditions of ambient temperature was studied by testing in a controlled-temperature furnace. The results were satisfactory. The frequency characteristic of the complete measuring system was shown to be determined by that of the receiver. Satisfactory methods of encapsulation and gauge bonding are illustrated.

26,573 **A New Gyrocompass for Smaller Vessels.** ARTHUR, R. J. *A.S.M.E., Paper No. 67–TRAN–52, presented* 28–30 Aug. 1967 [8 pp., 2 graphs, 8 diag., 2 phot.]

After outlining the history of the gyro-compass, the Author explains the basic design principles. He shows how a greatly simplified instrument of acceptable accuracy has recently been developed by eliminating all pendulous elements and using deck-plane azimuth. This instrument is described and illustrated; it is known as the Sperry Mark 27, and is sufficiently small, robust, and inexpensive for use on small vessels (a ship length of 50 ft is mentioned).

DECK MACHINERY AND CARGO HANDLING

26,574 **Air Cushion Pallet for Lifting Containers.** *Shipbuild. Shipp. Rec.*, 110 (1967), p. 846 (14 Dec.) [$\frac{1}{4}$ p.]

A short account is given of the Air Glide system for horizontal movement of containers aboard ship. The system is the result of a collaboration between the American Mail Line (the prospective users), the J. J. Henry Co. (their naval architects), and the Clark Equipment Co. (of Battle Creek, Michigan). The container to be moved is loaded on to a platform (also called a pallet) provided with air-cushion chambers on its underside. The cushion air is drawn from the ship's compressed-air system and " floats " the loaded platform on an air film about 0·0015 in thick. The platform is moved by a lift truck whose operator also regulates the air-supply pressure. During tests on an imitation deck, tilted at $4\frac{1}{2}°$, fully-loaded $20 \times 8 \times 8$ ft containers (22 tons gross weight) could be shifted about as desired; the time for a movement equivalent to that from hatch square to wing stowage in A.M.L.'s C5 cargo liners (now building) was under 60 sec. The system is to be used in these very versatile ships, which will be able to carry over 400 containers and, simultaneously, about 10,000 tons of bulk, palletised, or break-bulk cargo.

VIBRATION AND SOUND-PROOFING

(*See also Abstract No.* 26,572)

26,575 **Theory of Sound-Wave Propagation in Curved Wave-Guides (in Russian).** GRIGORYAN, F. E. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 81 (1967), p. 78 [7 pp., 4 ref., 2 tab., 3 graphs, 1 diag.]

The Author gives a wave theory for sound propagation in curved wave-guides (e.g. air-intake ducting) with rigid and with absorbent walls. The use of cylindrical co-ordinates leads to relatively simple formulae for wave number, etc.

CORROSION, FOULING, AND PREVENTION

26,576 **Potential Distribution in the Impressed-Current Cathodic Protection of Ships—Measurements on a Model Ship** (in German). LOHSE, U., HOLLNAGEL, M., and DOMMERDICH, H. *Schiffbautechnik*, 17 (1967), p. 249 (May) [4 pp., 5 ref., 5 graphs, 1 diag.]

In this investigation, point anodes surrounded by square insulating shields were disposed evenly over the model hull in accordance with the results of the work described in Abstract No. 24,092 (Mar. 1966). The model was floated in 3% NaCl solution; its cathodic regions were unpainted. Graphs show the potential distributions along the hull under various conditions. The effects of hull curvature, of electrolyte flow past the hull, of the anodes being closer to the water surface than to the keel, and of the presence of walls near the hull, are discussed. It was found, *inter alia*, that the current requirement increased rapidly with flow velocity up to about 1 ft/sec, but thereafter remained constant. The presence of a conducting or insulating wall had an observable influence on the potential distribution only when its distance from the hull was less than one-quarter of the anode separation.

OPERATION AND MAINTENANCE

26,577 **The Present State of the Reliability of Electrical Machines and Installations in the Polish Merchant Fleet** (in German). MARKIEWICZ, H. *Schiffbautechnik*, 17 (1967), p. 494 (Sept.) [4 pp., 7 ref., 2 tab., 3 graphs]

In introducing this German abridgement based on a paper in Polish presented at Gdansk in Sept. 1966, the writer (H. Gröbe, of Rostock University) mentions that electrical failures similar to those reported in Polish ships have occurred in East German ships, and that, particularly in view of the increased use of automation, the information given should be of considerable interest in East Germany.

Tables are presented showing (i) the total electrical power installed, for each of the usual applications, in the Polish merchant fleet, and (ii) the number of D.C. and A.C. generators installed for each power rating (there are 51 D.C. ratings, ranging from 3·2 to 360 kW, and 22 A.C. ratings, ranging from 30 to 500 kW). The 73 different power ratings listed in (ii) embrace more than that number of generator types because of variations in voltage and r.p.m., though the total number of generators is only 580.

Column graphs, applying to about half the sea-going merchant fleet (excluding coasters and fishing vessels), show the 1964 distributions of D.C. and A.C. motors on a basis of power per motor. Another shows, on the same basis and with a rudimentary analysis by motor function, the distribution of service failures in these motors; only failures which could not be rectified by the crew are included. This distribution of failures is discussed.

General information, together with a few statistics, is given on generator failures which needed rectification by the shipyard or by the manufacturers (the generators concerned were often of non-Polish manufacture). Some information is also given on difficulties experienced with electrical equipment other than motors and generators, and data are given on faults, in electrical equipment in general, rectified by shipboard personnel.

It should be possible to considerably reduce the excessive numbers of faults and failures, and the amount of maintenance work required, by improving the reliability of electrical equipment and installations. There is no doubt that improved reliability, together with better retention of performance with age, is the most pressing problem in marine electrical technology as far as Poland is concerned. The financial consequences of poor reliability are briefly discussed.

The idea that electrical equipment designed for land use can, by simple modification, always be made suitable for ships must be revised. Classification Rules are out-of-date in many respects, and should be amended to cover long-term reliability. The article concludes with some views (mainly concerning organisation) on how marine electrical technology in Poland can be brought up to a higher standard.

26,578 The Application of General Methods of Reliability Theory to Analysis of the Maintenance of Marine Diesel Engines (in Russian). VINOGRADOV, V. I. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 81 (1967), p. 25 [12 pp., 13 ref., 7 tab.]

Statistical principles are given for the selection of optimum intervals for maintenance operations on main Diesel engines. A theoretical basis is proposed for the classification of components according to the degree of risk of failure. Examples are given of the preliminary analysis of service data for fuel pumps and engine bearings in existing motor ships.

26,579 " In Situ " Honing and Machining of Worn Bearing [Journal] Surfaces. *Motor Ship*, 48 (1967), p. 359 (Nov.) [1 p., 1 tab., 2 phot.]

Nicol & Andrew Ltd, of London and Glasgow, are specialists in this field and concentrate on marine work. The article describes the portable equipment developed and used by them; this equipment is suitable for many marine applications, including tailshafts (up to 32 inches in diameter) and the journals of large crankshafts. It has two main parts: a transmitting (power) unit, and a split head unit which is assembled around the shaft or journal to be treated. The transmitting unit is in two separable parts, one incorporating an electric motor and the other a gearbox which is driven by the motor through V-belts and itself drives, through a universal coupling, the output pulley for the transmission belt. A flat Balata belt is used to ensure slipping in the event of overload on the head, and an equalising device is fitted to maintain uniform pressure around the workpiece. The head unit has a peripheral groove for the belt, and a

frame with thread-type adjusters to compensate for stock removal and stone wear. The frame has mountings for four stone-shoe carriers; as these are not spring-loaded, the stones do not follow irregularities in workpiece form. For rapid stock removal ($\frac{1}{16}$ in or more) a special tooling head has been developed; this is a split carriage of adjustable length, with a traversing tool slide. (In the case of a crankpin the fillet radii are used as tracks.)

Some details are given of a particular job, *viz* reconditioning the crankshaft of a six-cylinder Doxford propulsion engine which had been under water for about a fortnight in the flooded engine room of an 18,000-ton d.w. tanker. The crankshaft was extensively corroded, with pitting to a depth of 0·02 in on the centre crankpins. Four technicians and three machines were flown from Glasgow to Singapore, and the job (which involved machining 0·04 in from the diameter of six centre crankpins, and honing seven main journals and 12 side crankpins) was completed in 14 days.

26,580 **Type B-15 Fire-Classified Cabin-Door** (in German). BOECKEL, B. *Schiffbautechnik*, 17 (1967), p. 703 (Dec.) [3 pp., 1 tab., 3 diag., 3 phot.]

In East Germany, as elsewhere, the production of a satisfactory " B " Class cabin-door conforming to the 1960 Safety Convention has presented difficulties. However, the B-15 cabin-door developed by the VEB Isolier- & Kältetechnik, Rostock, conforms to the Convention requirements and is aesthetically suitable for passenger ships.

The leaf of this type of door is an asbestos-fibre panel 24 mm (0·94 in) thick; this is faced on each side with Melamine laminated sheet 1 mm thick. The panel edges are reinforced with specially-shaped projecting steel sections of 1 mm material, which constitute a framing around the panel. Legs of the inside and outside sections overlap in the thickness direction, but are separated by a 3 mm (0·12 in) asbestos strip so that there is no metal-to-metal contact. This framing, in addition to its structural function, forms the bearing surface for the door closure. In the lower half of the door panel are a ventilator and an emergency knock-out panel. The article gives further information on the construction of the door and its fittings, together with drawings. The principle of separation of overlapping metal parts by insulating strips was basic to the design.

A description is given of a furnace test on the door and of a separate furnace-test on the door-lock and adjacent area. Numerical and other results are given and briefly discussed. Following these tests, the design received DSRK approval.

The door is made by the " central producer " of cabin doors, the Neptun shipyard at Rostock, which supplies it to other East German yards. Since 1966, the door has been installed in all seagoing ships built by these yards, including ships for other countries. Patents, both East German and foreign, have been applied for.

MISCELLANEOUS

26,581 **The Port of the Future.** JOHNSON, S. *Instn E. Shipb. Scot., Paper No. 1328, presented* 16 Jan. 1968 [14 pp., 6 ref., 3 tab., 1 graph]

The Author discusses the factors which will govern the type, location,

and capacity of U.K. ports in the immediate future, under the main headings: present physical characteristics and limitations; growth of world trade; technological changes (especially containerisation); and port organisation.

26,582 An Approach to the Shipboard Icing Problem. LANDY, M., and FREIBERGER, A. *Naval Engrs J.*, 80 (1968), p. 63 (Feb.) [9½ pp., 23 ref., 3 tab., 3 graphs, 1 diag., 2 phot.]

The main object of the comprehensive investigation of ice adhesion here described was to establish criteria and performance standards for low-adhesion coatings that can be applied to weather surfaces and thus obviate the laborious manual methods at present used on ships for removing ice accretions.

The Authors first examine the manner in which ice adheres to a substrate, and the properties of the substrate and of the ice that affect ice removal. In an earlier paper they showed that the adhesion of ice to a surface is caused by a chemical reaction that occurs between the surface and the water before it freezes. They showed also that the ice adhesion increases with the flexural modulus of the substrate and, for a given material, with the thickness of the specimen.

The nature of the bond between ice and substrate is explained, and methods of measuring the strength of the adhesion bond are discussed. The measured values obtained by different test methods differ, and in any case depend on the test conditions. Among the latter, water purity, rate of freezing, interfacial area, height of ice block, test temperature, age of bond, and rate of stress application are discussed. The effects of some of these variables are shown as graphs; for the others, references are given to published literature.

To determine whether the adhesion values as determined in the laboratory bear any relation to the ease or difficulty of ice removal aboard ship, the Authors devised a drop-ball test to determine the amount of ice removed when nine equally-spaced steel balls weighing about 1·2 lb each are dropped in rapid succession from a height of 2 ft on to a mild-steel panel 1 ft square by ½ in thick covered with about ¼ in of ice. The amounts were found to bear an approximately linear inverse relationship to the shear strength of the adhesive bond.

A table shows the results obtained of the measured ice adhesion strength of a number of coatings on painted steel. The Dow Corning compound F-121 gave the lowest ice adhesion but the results showed large variations from lot to lot. Excellent reproducibility was shown by an experimental silicone resin XZ-8-3057, also by Dow Corning, which gave adhesion values not much higher than those of F-121. In service trials aboard ship, XZ-8-3057 reduced the amount of work required to remove ice by up to 70%. It is being considered for interim acceptance by the U.S. Navy. It has the disadvantages of a service life of only about two weeks, of low abrasion resistance, and of being slippery when wet so that it cannot be used on walkway surfaces.

Another material with good de-icing properties is Teflon FEP. Because of its high flexibility it can be wound around ships' rigging, masts, and aerial lines.

26,583 **Dynamics of a Floating Ice Sheet.** REISMANN, H., and LEE, Y.-C. *Journal of Hydronautics*, 2 (1968), p. 108 (Apr.) [4 pp., 7 ref., 1 tab., 7 graphs]

This dynamic analysis considers the deformation of a floating ice sheet of infinite extent subjected to rapidly applied surface loads.

26,584 **Buoy Techniques for Obtaining Directional Wave Spectra.** CARTWRIGHT, D. E., and SMITH, N. D. Reprint from *Buoy Technology* (*Marine Technology Society, Washington*), p. 112 (1964) [10 pp., 8 ref., 3 graphs, 2 diag.]

The paper describes wave-measuring buoys and buoy systems recently developed by the National Institute of Oceanography with the object of improving directional resolution. Accounts are given of improved versions of the " Pitch-Roll " buoy, of the " Cloverleaf " three-float buoy (which is shown in diagrams), and of a system comprising a line of " Pitch-Roll " buoys towed very slowly by a ship. In each case the underlying theory is explained. Some results of comparative tests of the Cloverleaf buoy and the line of buoys are presented; the former is shown to be superior.

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